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Genter et al.

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(54) **APPARATUS AND METHOD FOR ADJUSTING A GEAR**

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(73) Assignee: **Cummins Engine Company, Inc.**, Columbus, IN (US)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

This patent is subject to a terminal disclaimer.

(21) Appl. No.: **09/489,755**

(22) Filed: **Jan. 21, 2000**

Related U.S. Application Data

(63) Continuation of application No. 09/186,238, filed on Nov. 4, 1998, now Pat. No. 6,109,129, which is a continuation-in-part of application No. 08/853,341, filed on May 8, 1997, now Pat. No. 5,870,928, and a continuation-in-part of application No. 08/853,013, filed on May 8, 1997, now Pat. No. 5,979,259.

(51) **Int. Cl.**[7] **F16H 55/18**
(52) **U.S. Cl.** **74/440**; 74/397; 74/409
(58) **Field of Search** 74/397, 409, 440

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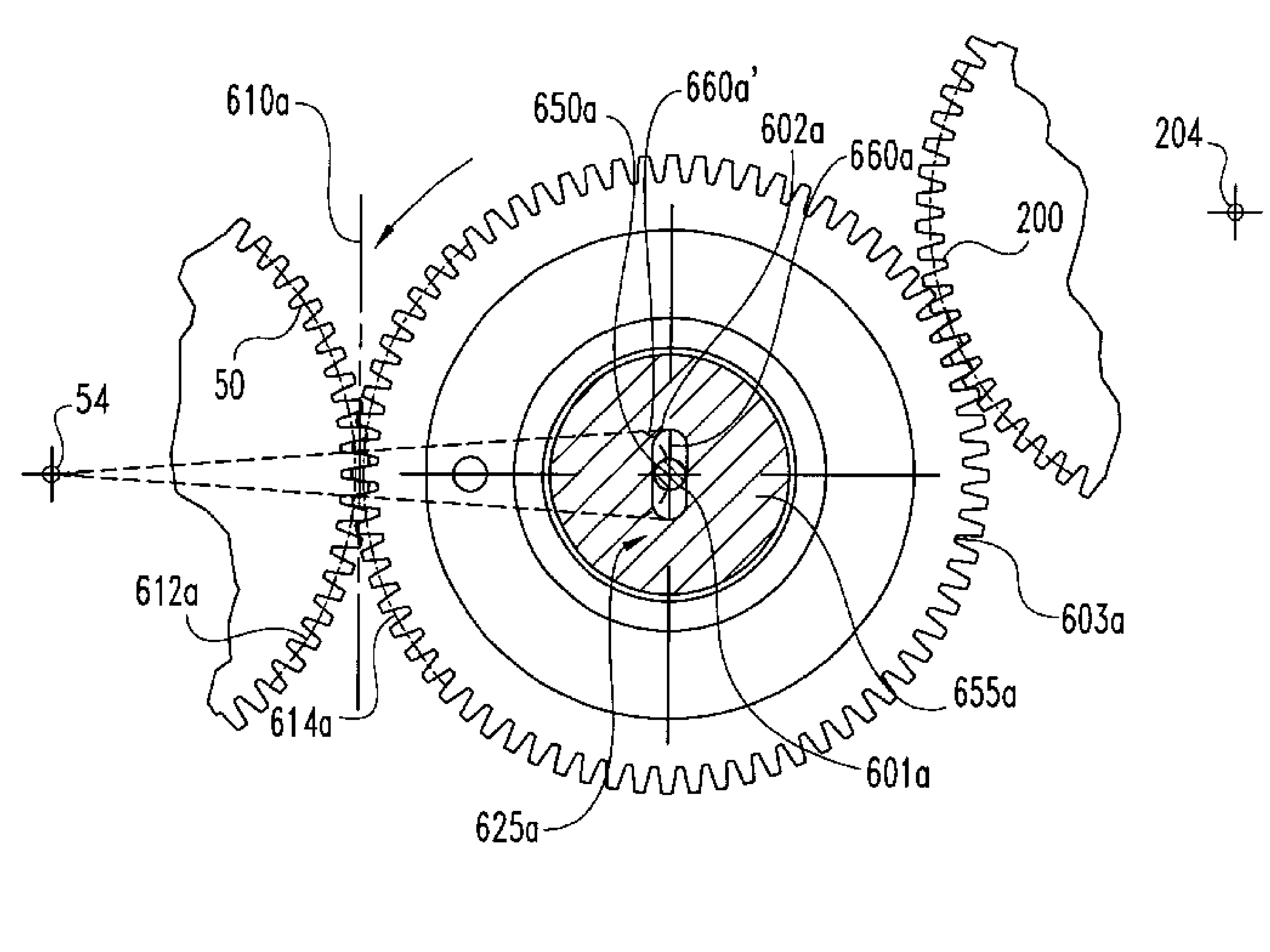
(57) **ABSTRACT**

Apparatus and method for adjusting a gear in a gear train assembly. A first gear with a first rotational center and a second gear with a second rotational center are coupled to an engine. A third gear with a third rotational center is provided to form a first mesh with the first gear. The first mesh has a predetermined minimum backlash. The third gear also forms a second mesh with the second gear. The third gear includes an adjustable positioning mechanism which allows the rotational center of the third gear to be adjusted along a predetermined adjustment path that is substantially tangent to the first mesh. The adjustment mechanism of the third gear allows the minimum backlash of the first mesh to be maintained while the second mesh is adjusted to likewise achieve a generally minimized backlash.

10 Claims, 21 Drawing Sheets



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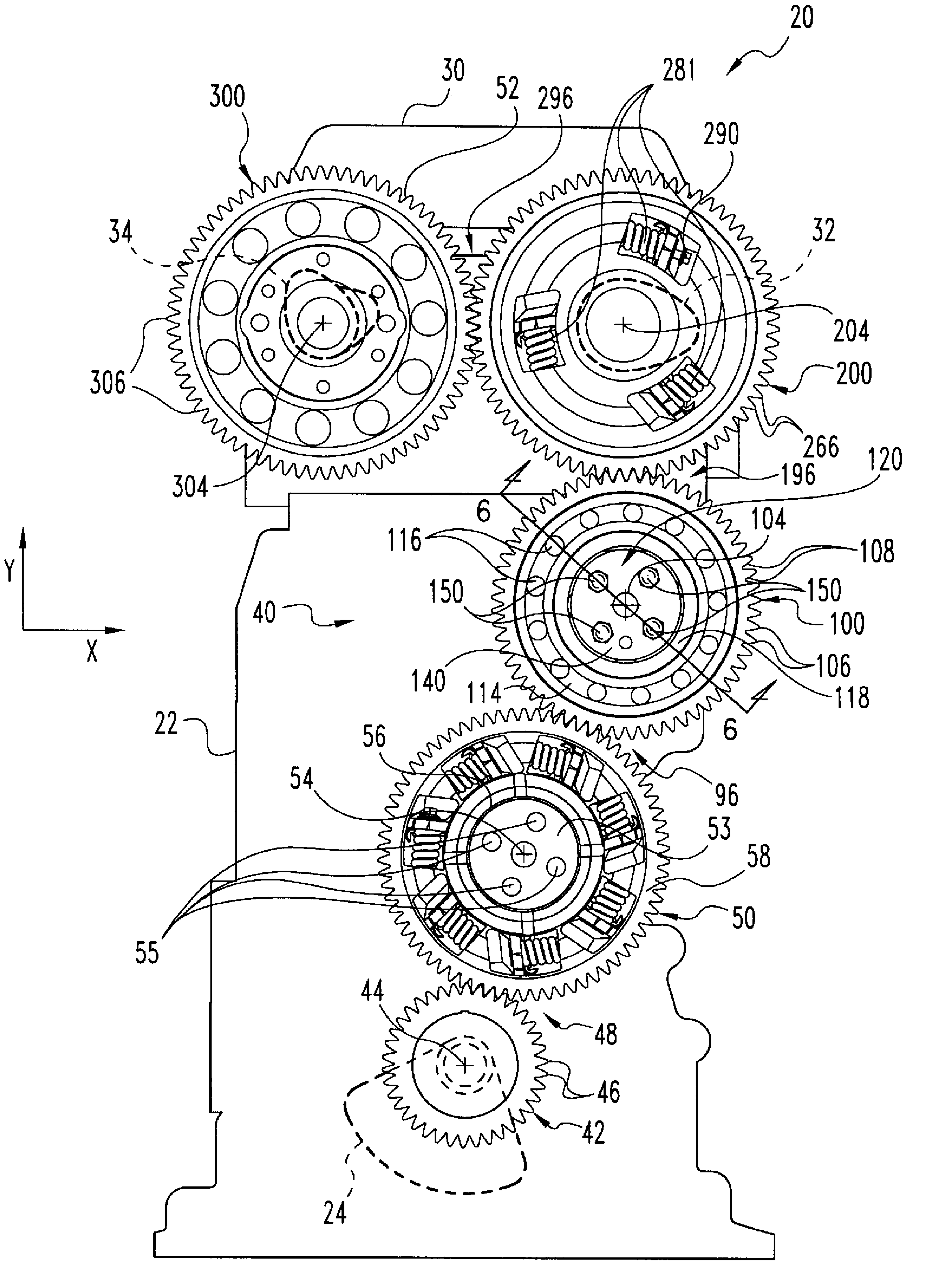
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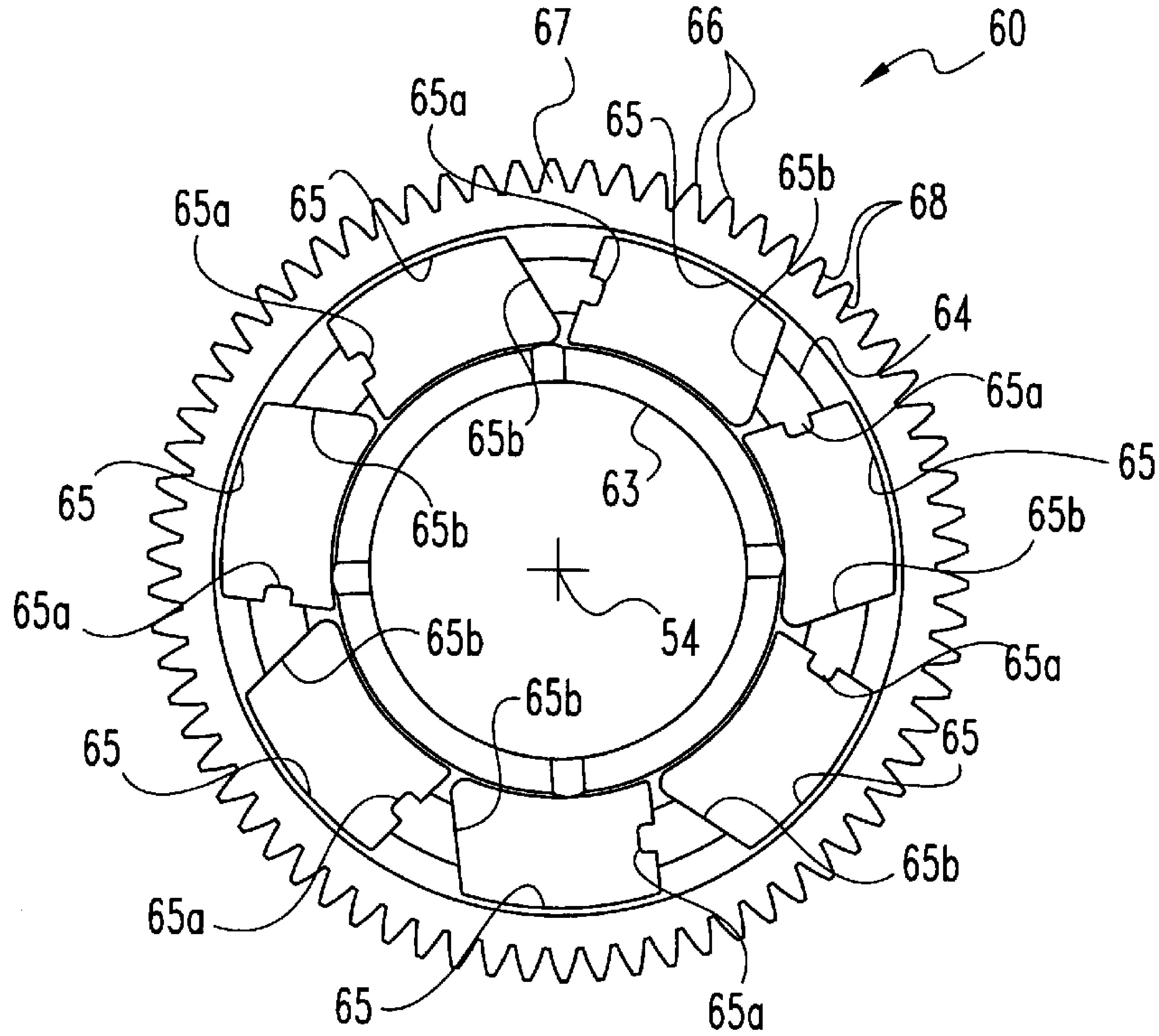
Y
150
150
X
40
100
140
114
106
118
22
6
56
96
54
53
58
55
50
44
48
46
42
24

Fig. 1



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67
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65a
65
65b
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65a
65
64
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65b
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54
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65b
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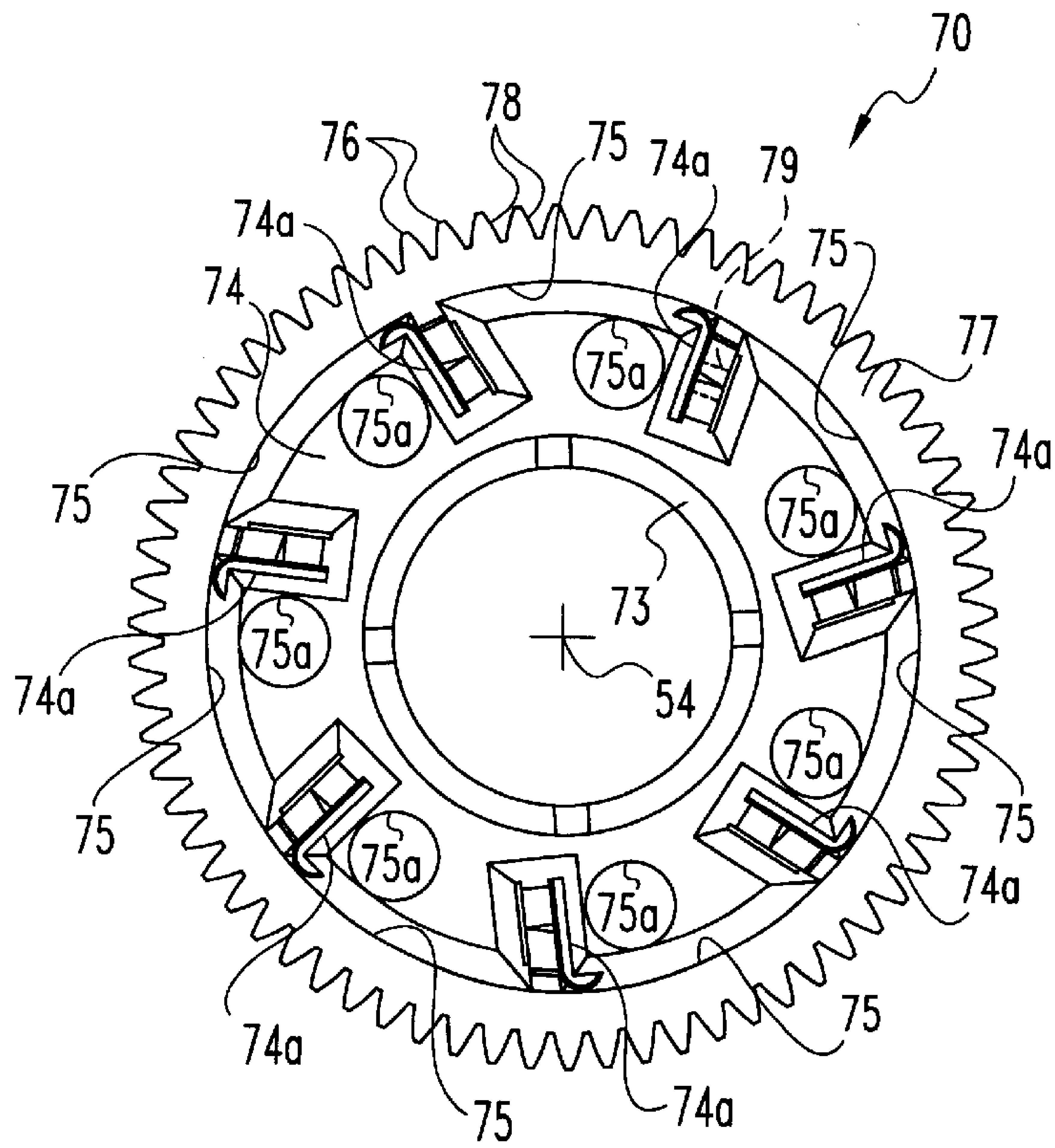
65
65
65b
65a
65
65a

Fig. 2



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74a
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74
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74a
75a
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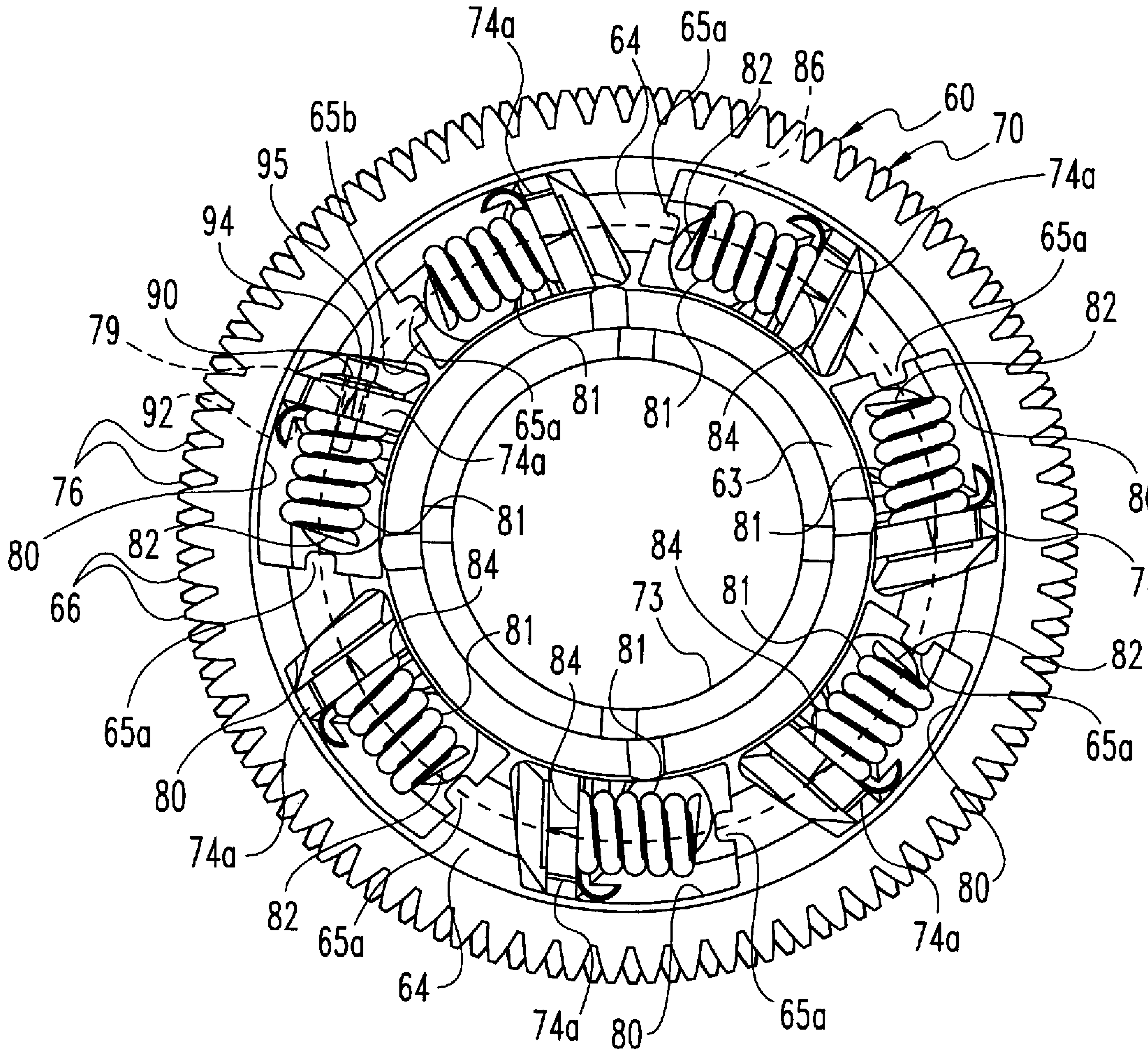
75
74a
75

Fig. 3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65a
80
74a
82
65a
64
74a
80
65a
74a
80
65a
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74a
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84
73
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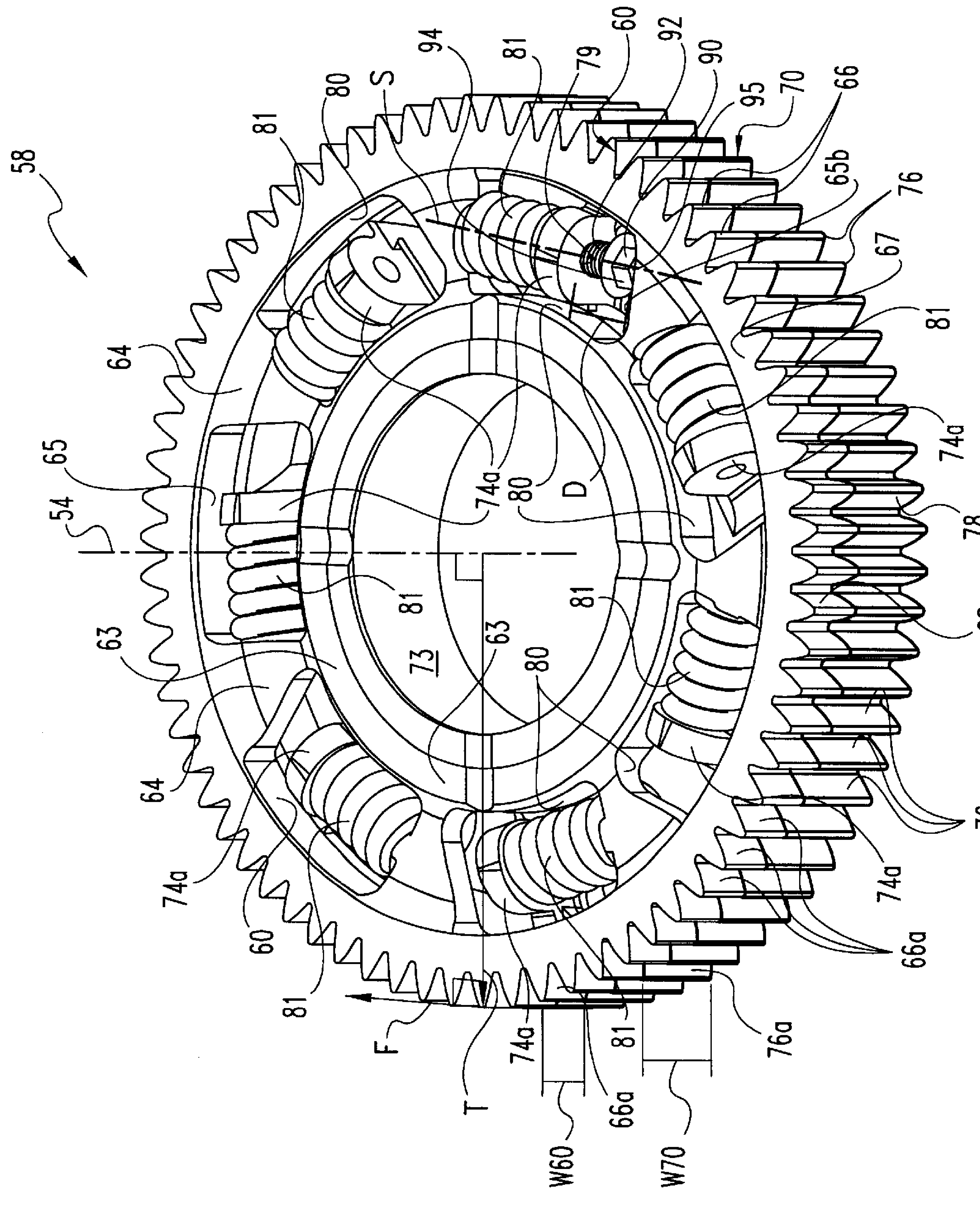
81
84
81

Fig. 4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76a
68
78
74a
81
67
76
65b
66
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95
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60
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81
94
S
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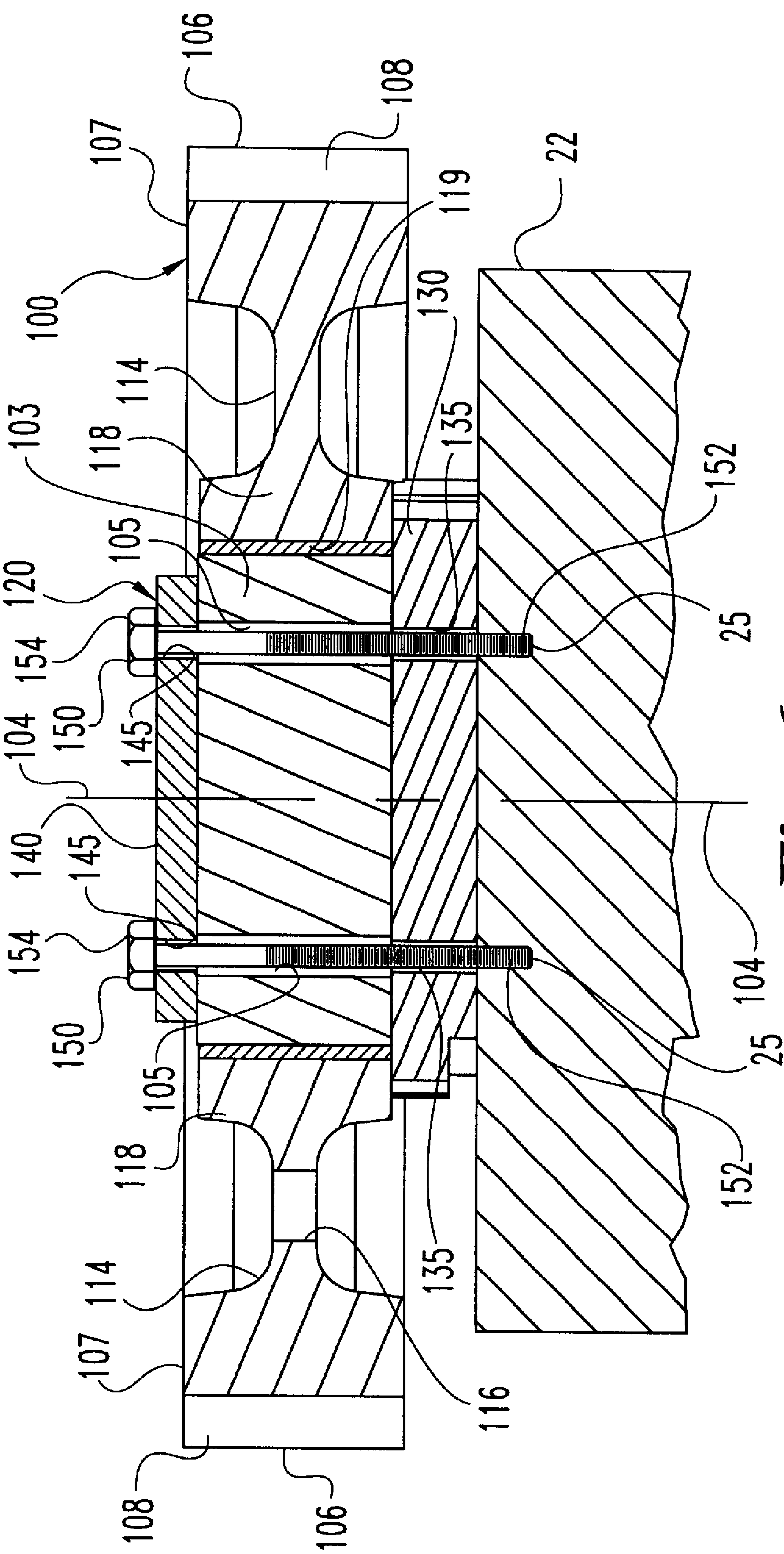
81
74a
80
D
81
73
63
80
81

Fig. 5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152
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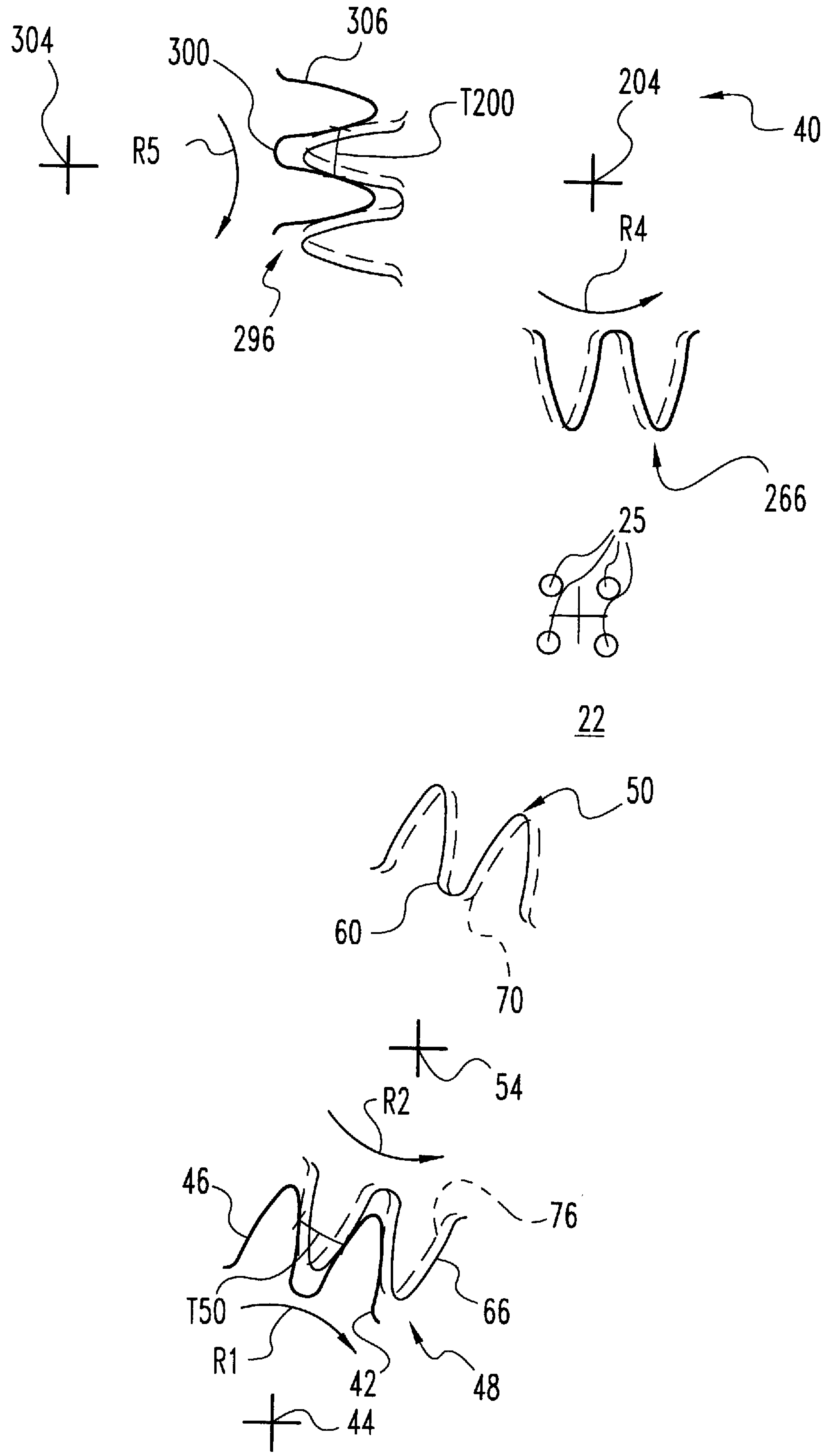
116
108
106

Fig. 6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204
40
R5
R4
296
266
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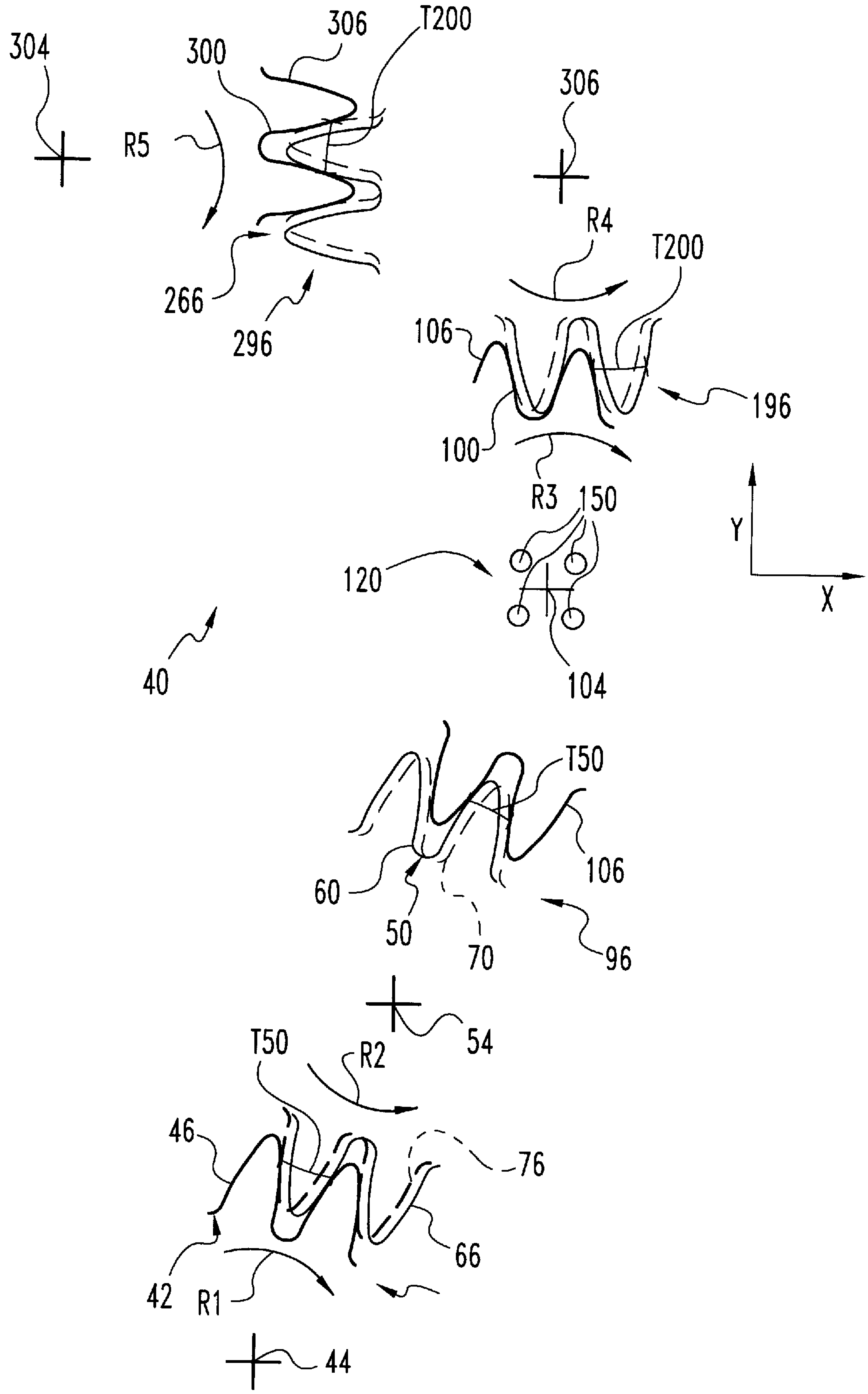
50
60
70
54
R2
46
76
T50
66
R1
42
48
44

Fig. 7A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T200
R5
306
266
296
R4
T200
106
196
100
R3
150
Y
X
120
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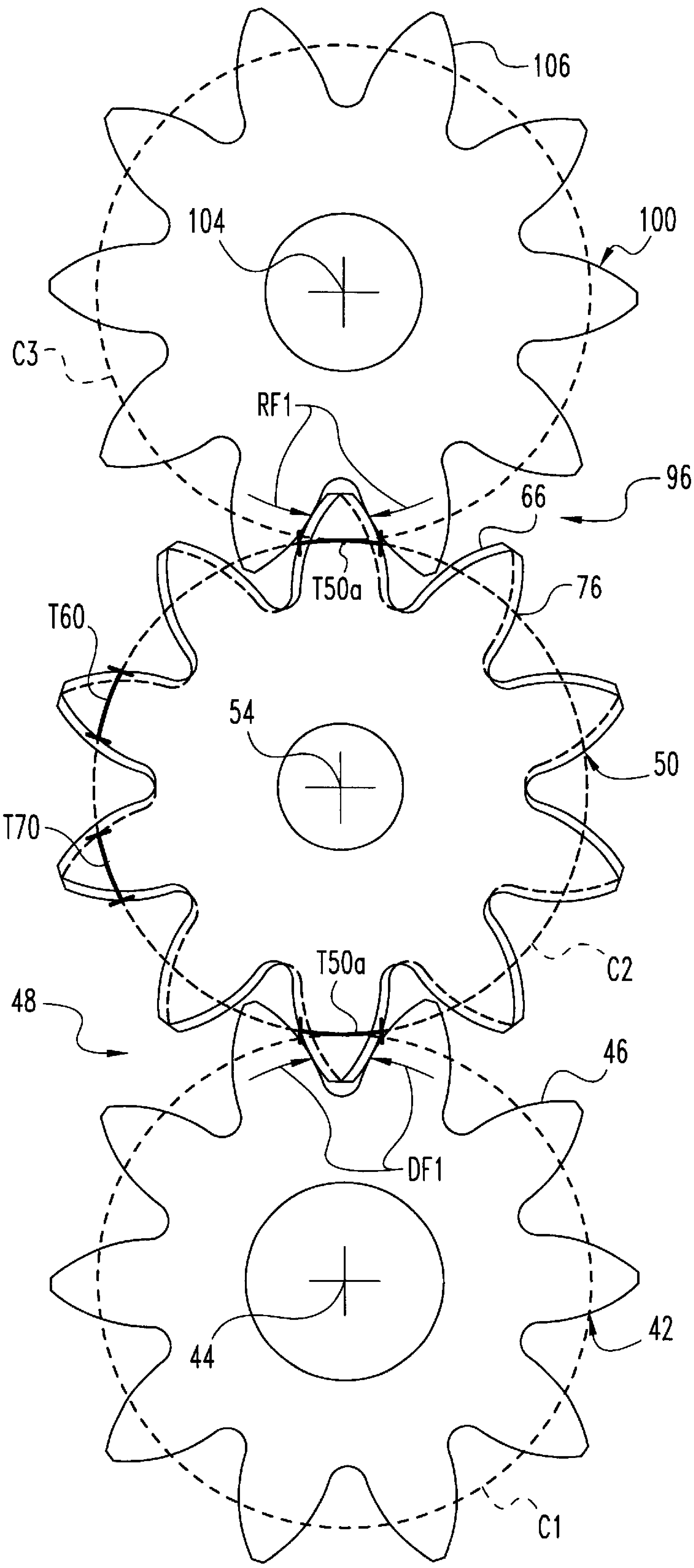
104
T50
106
60
50
70
96
54
T50
R2
46
76
66
42
R1
44

Fig. 7B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100
C3
RF1
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66
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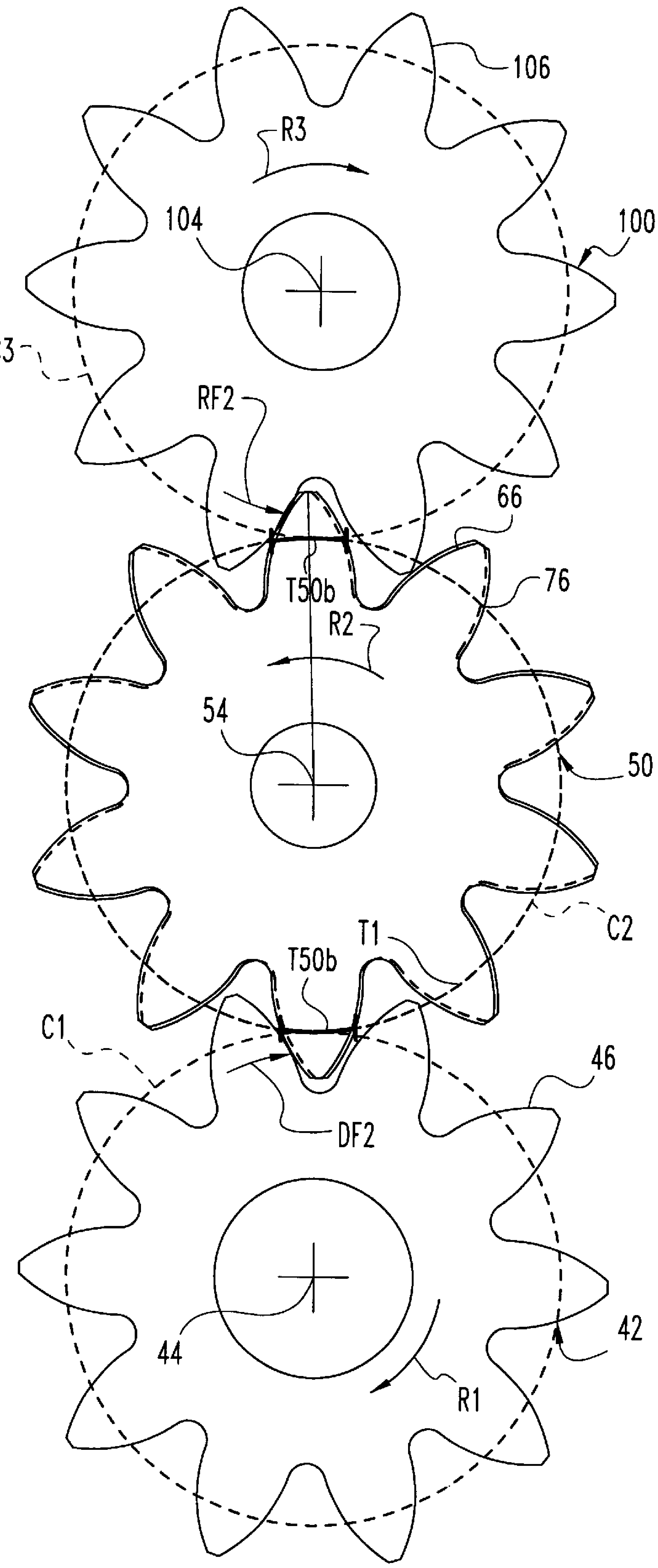
76
T60
54
50
T70
C2
T50a
48
46
DF1
42
44
C1

Fig. 8A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R3
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C3
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50
C2
T1
T50b
C1
46
DF2
44
42
R1

Fig. 8B

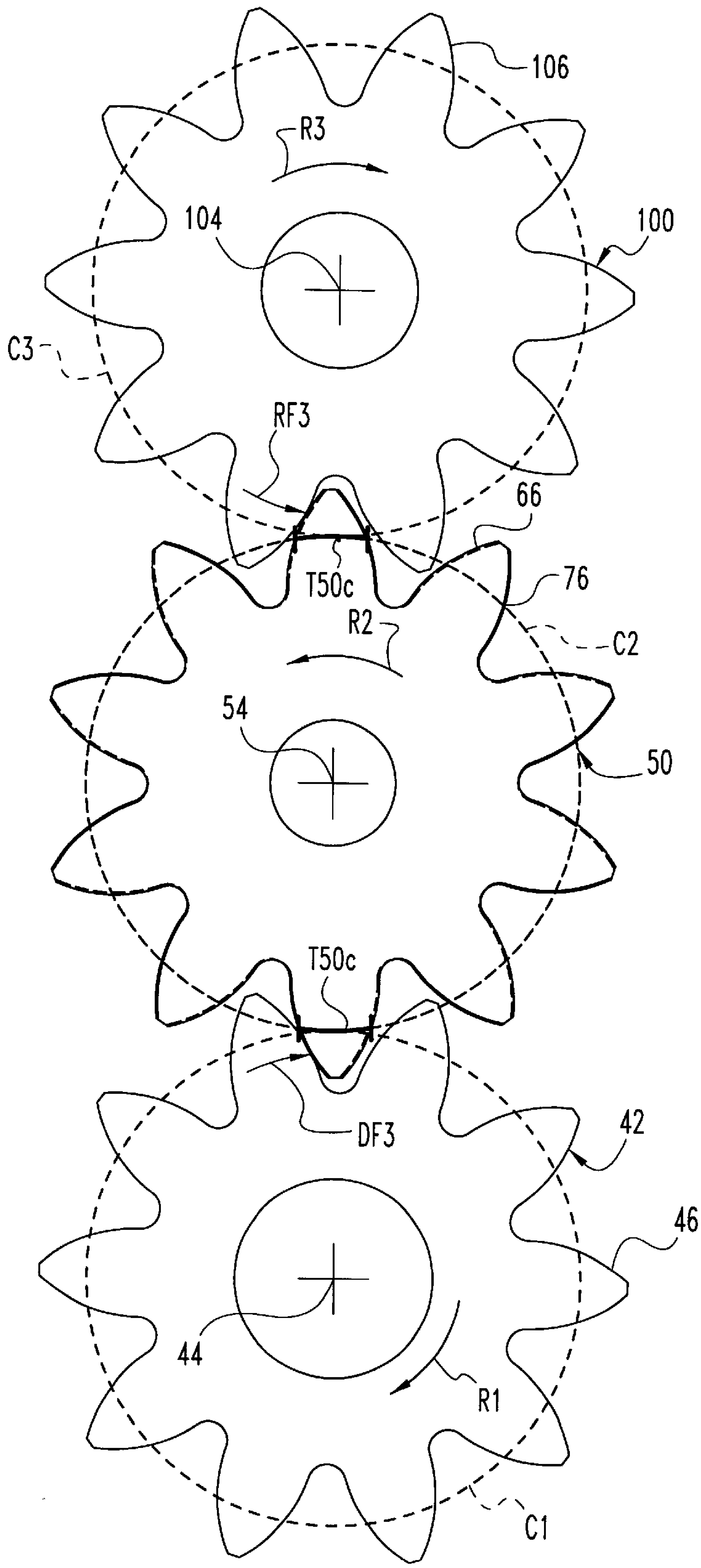
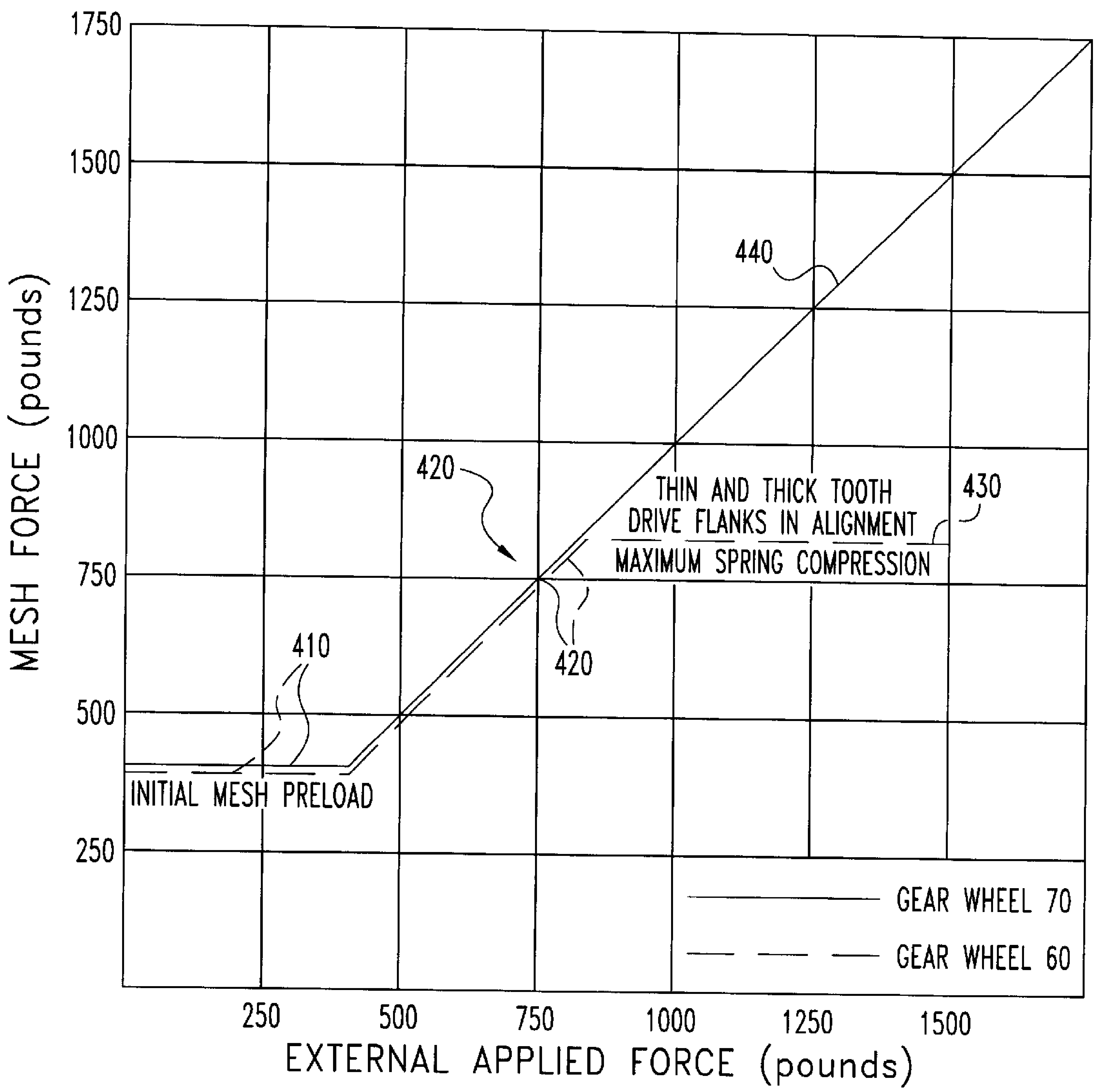
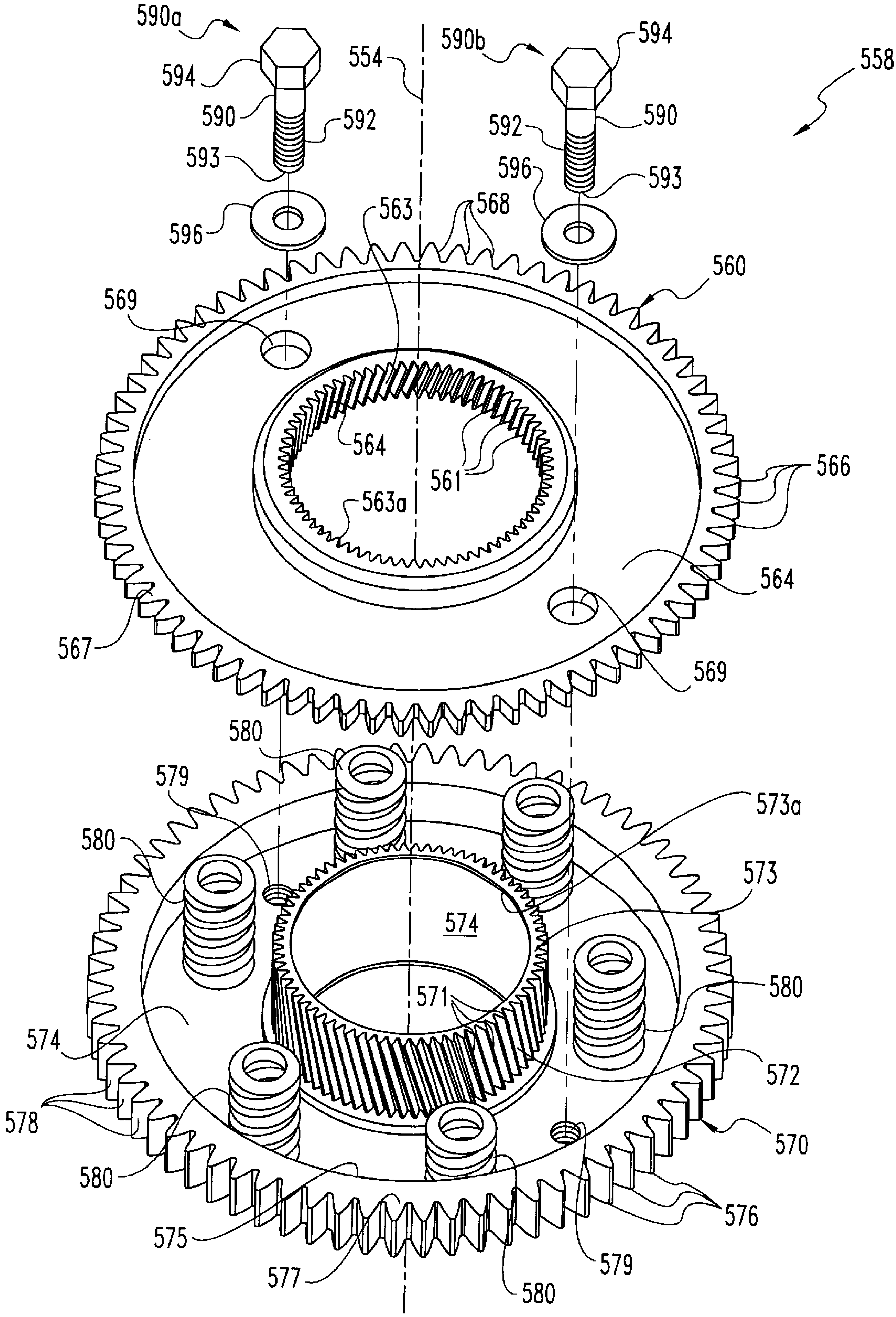


Fig. 8C



MESH FORCE (pounds)
1750
1500
1250
1000
750
500
250
EXTERNAL APPLIED FORCE (pounds)
250
500
750
1000
1250
1500
440
420
THIN AND THICK TOOTH DRIVE FLANKS IN ALIGNMENT
430
MAXIMUM SPRING COMPRESSION
420
410
INITIAL MESH PRELOAD
GEAR WHEEL 70
GEAR WHEEL 60

Fig. 9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590b
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566
563a
564
567
569
580
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573a
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579
576

Fig. 10

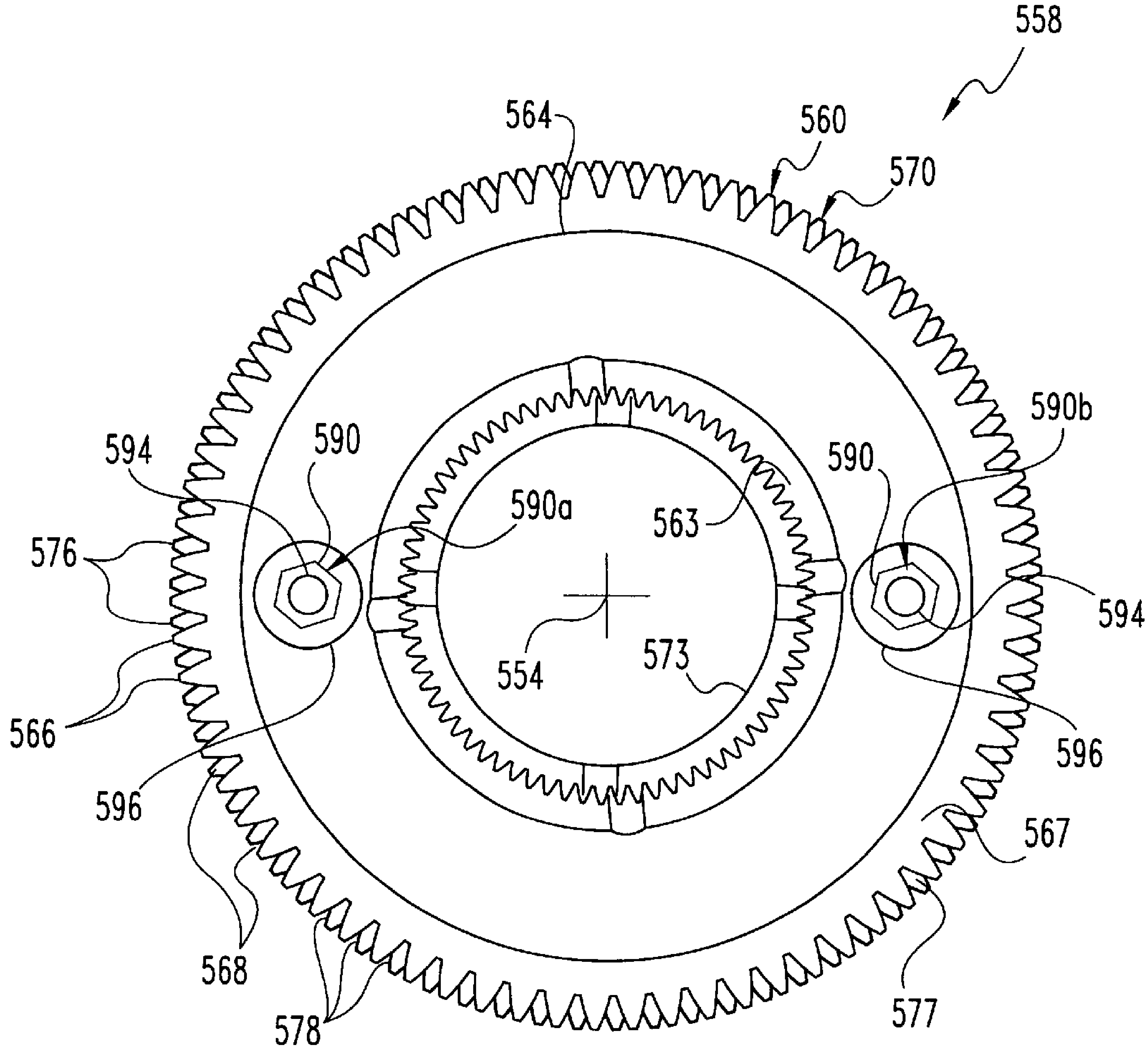
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Fig. 11A

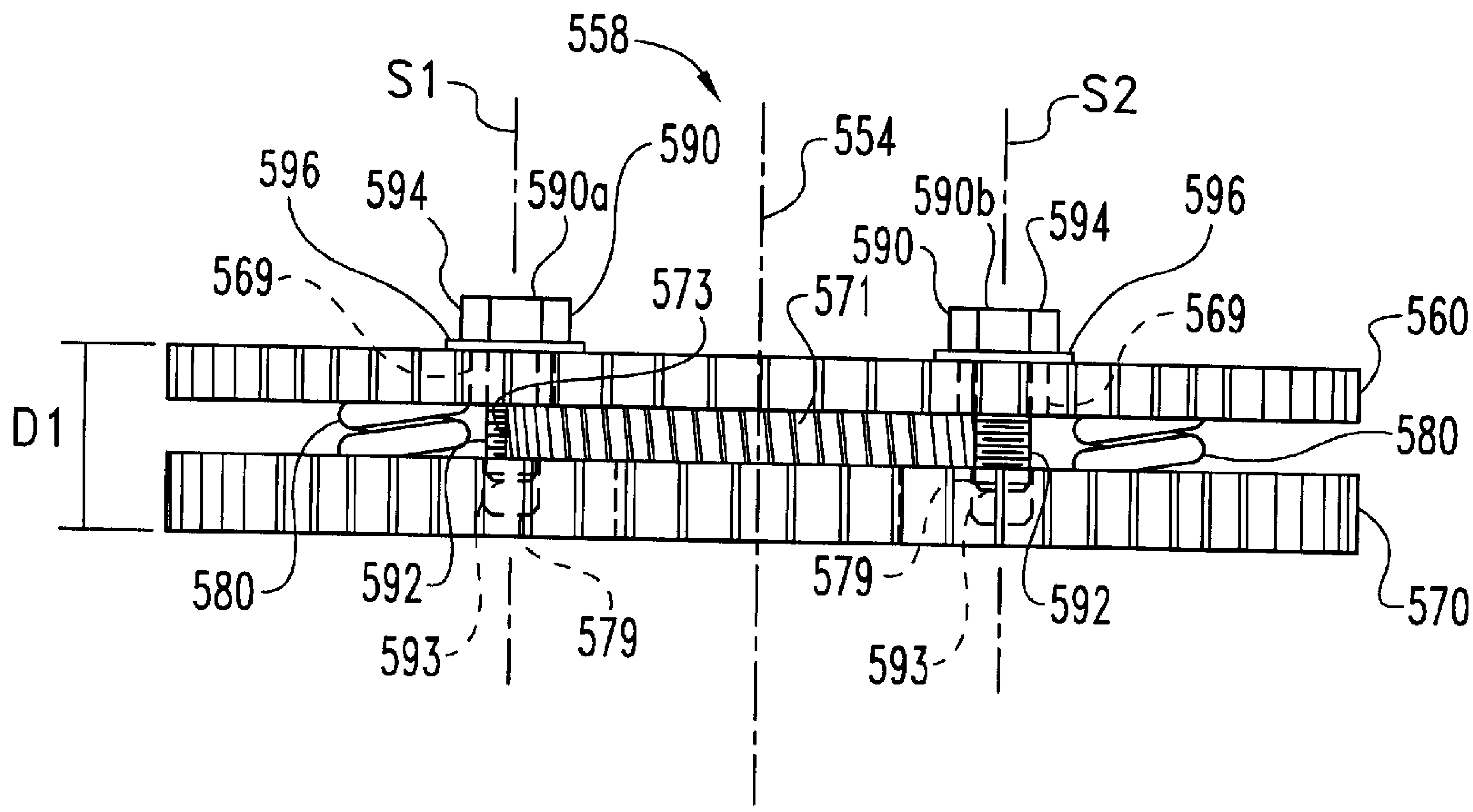


Fig. 11B

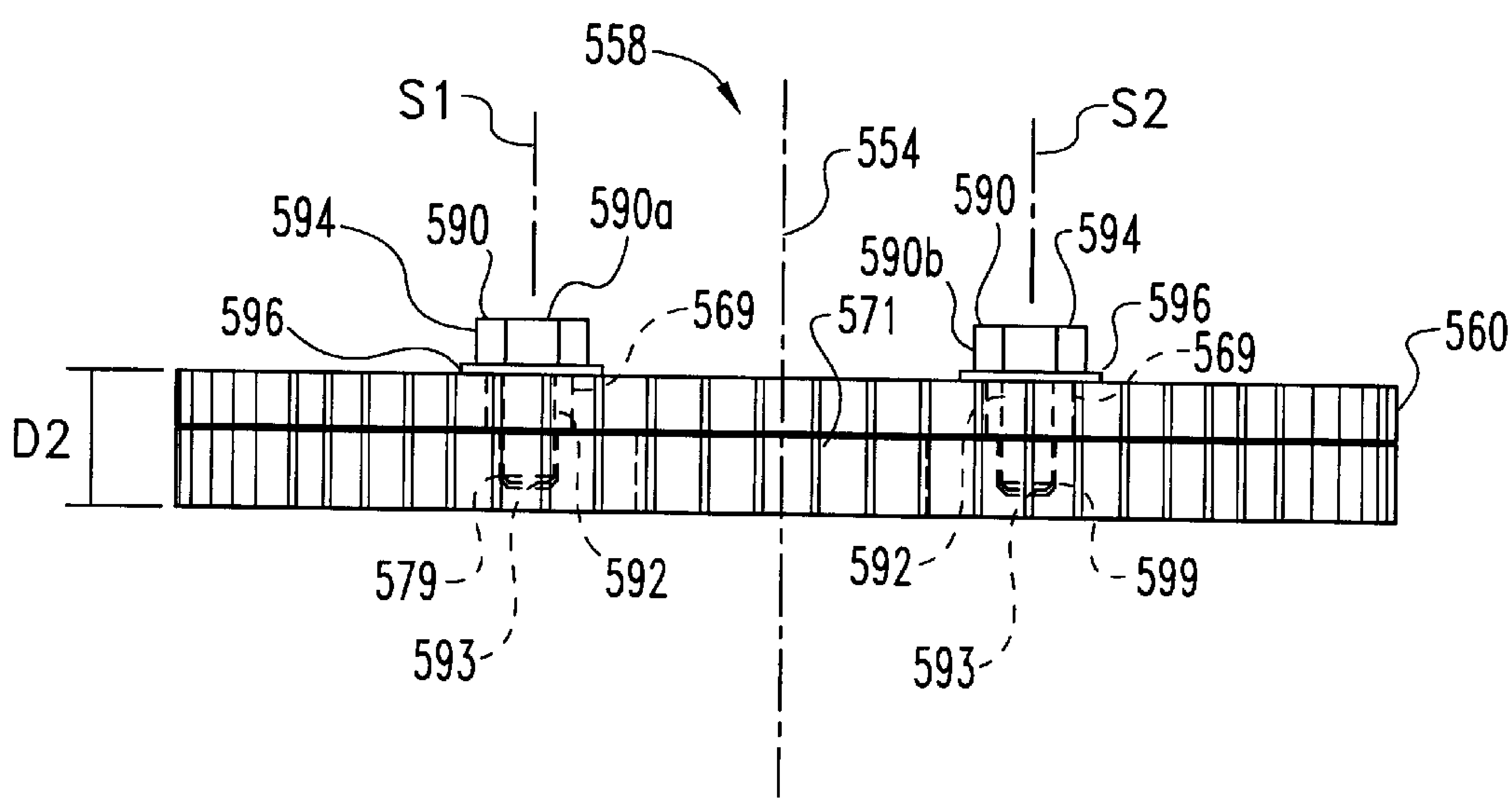
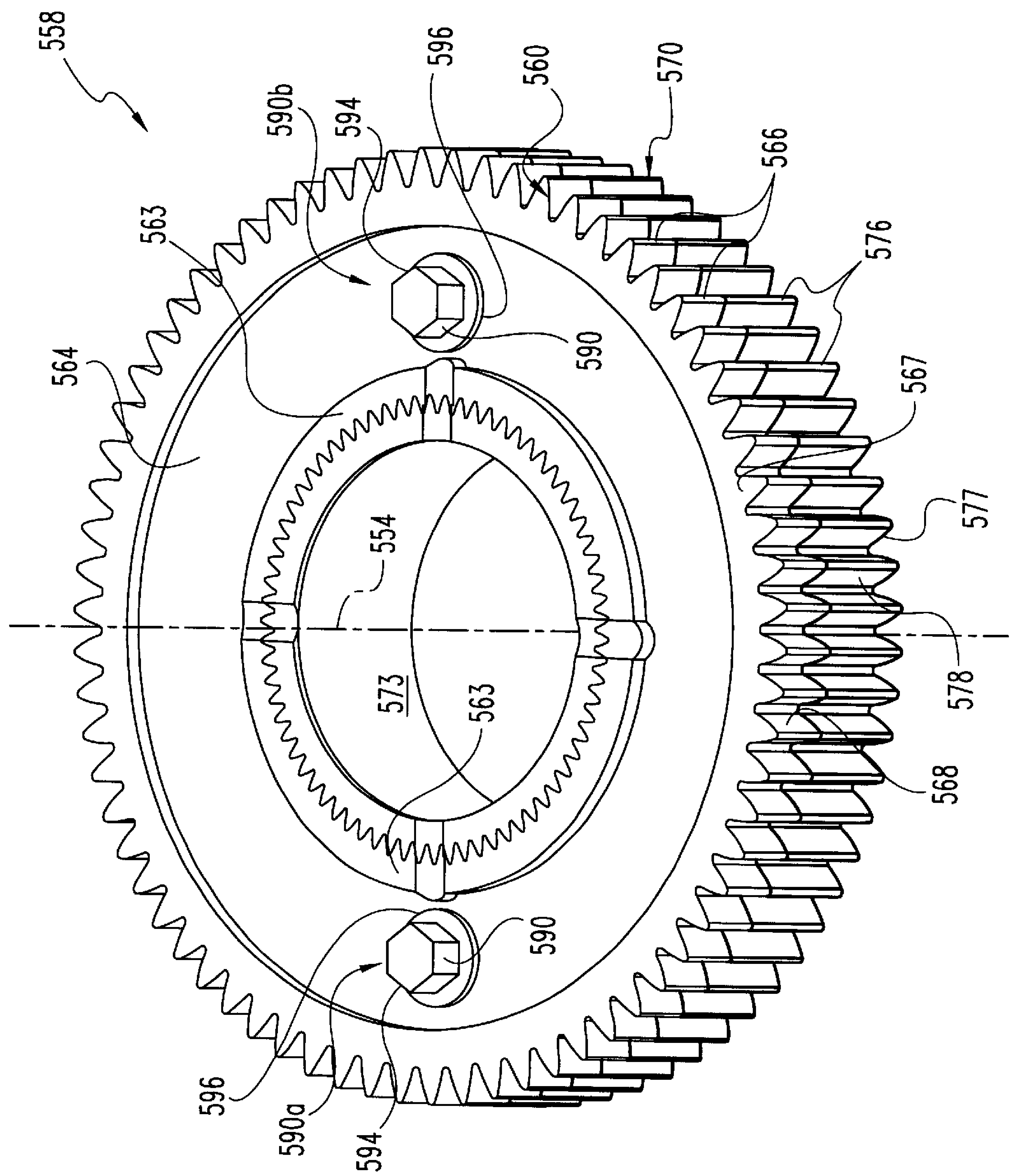


Fig. 12B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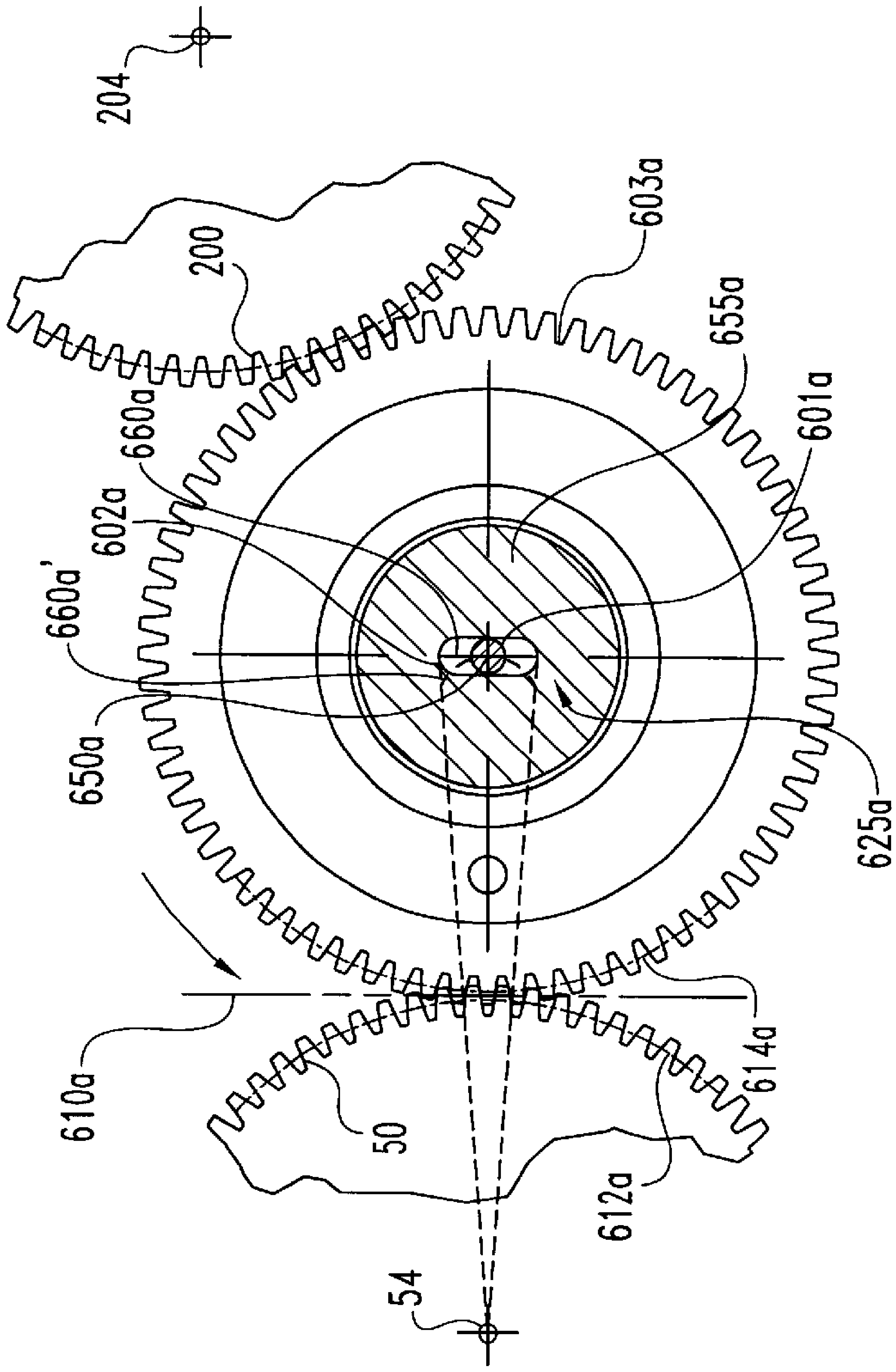
590a
594

Fig. 12A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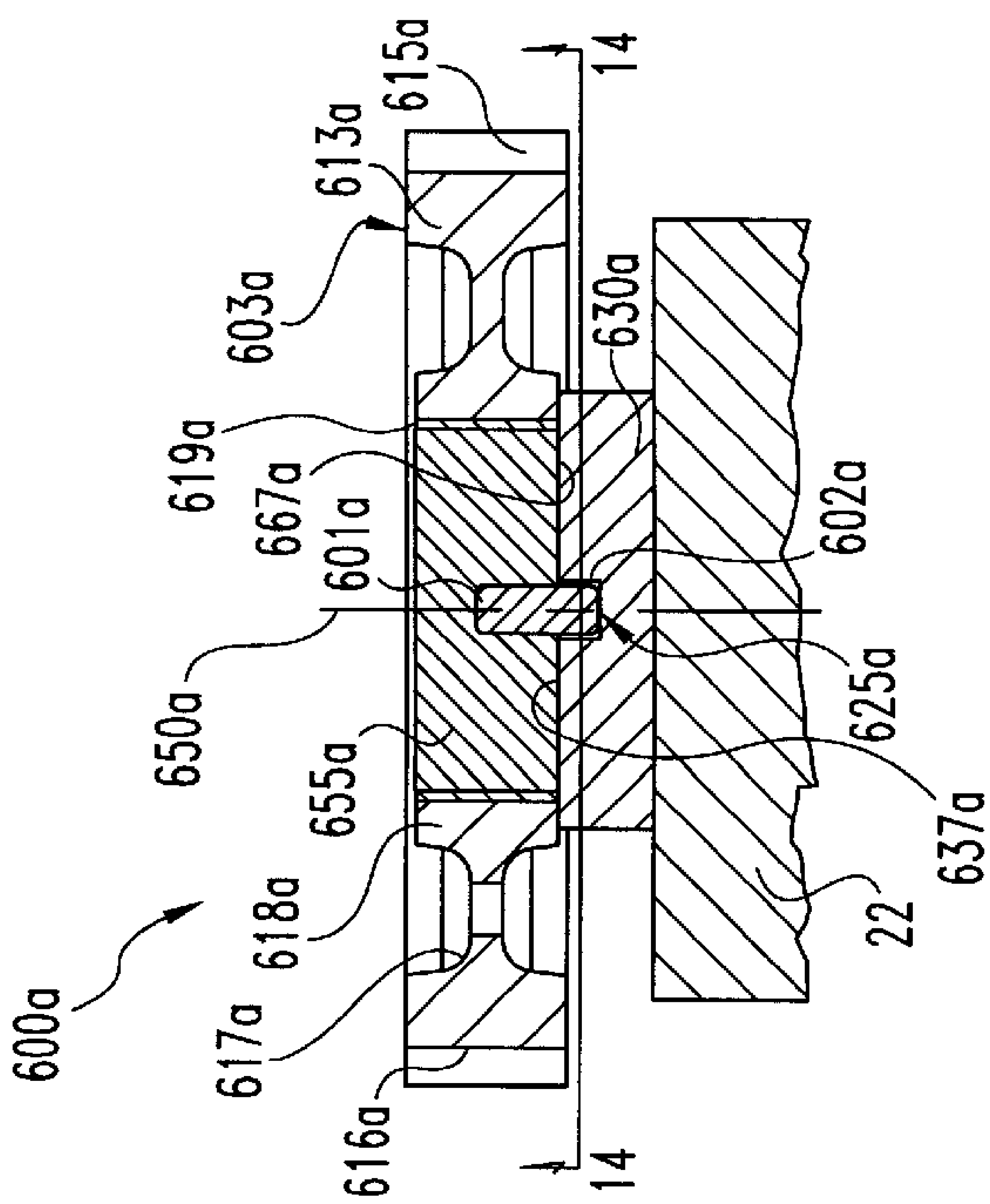
660a
602a
660a'
650a
625a
610a
50
614a
612a
54

Fig. 14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603a
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619a
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602a
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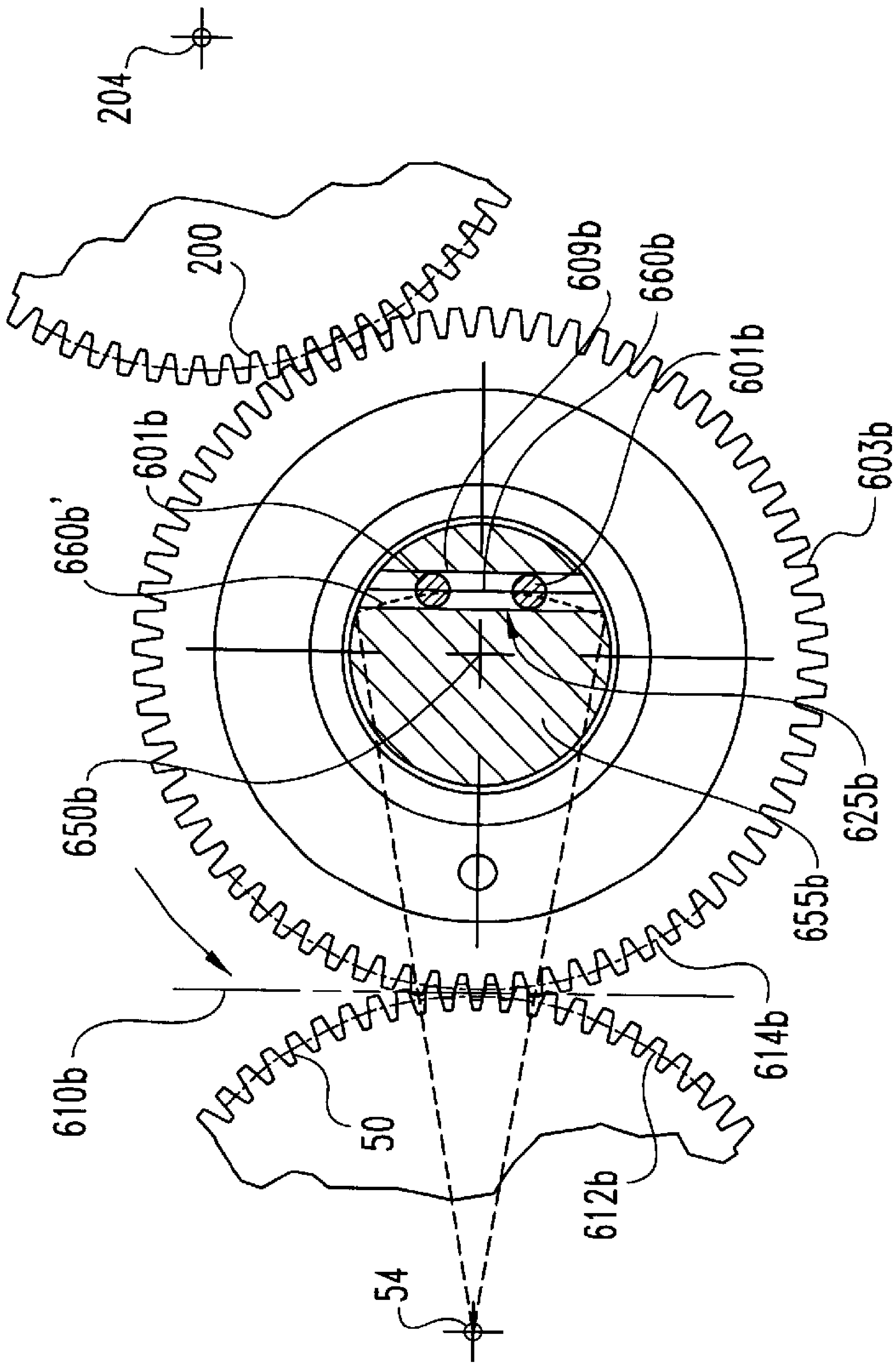
625a
655a
637a
618a
22
600a
617a
616a
14

Fig. 13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601b
601b
603b
660b'
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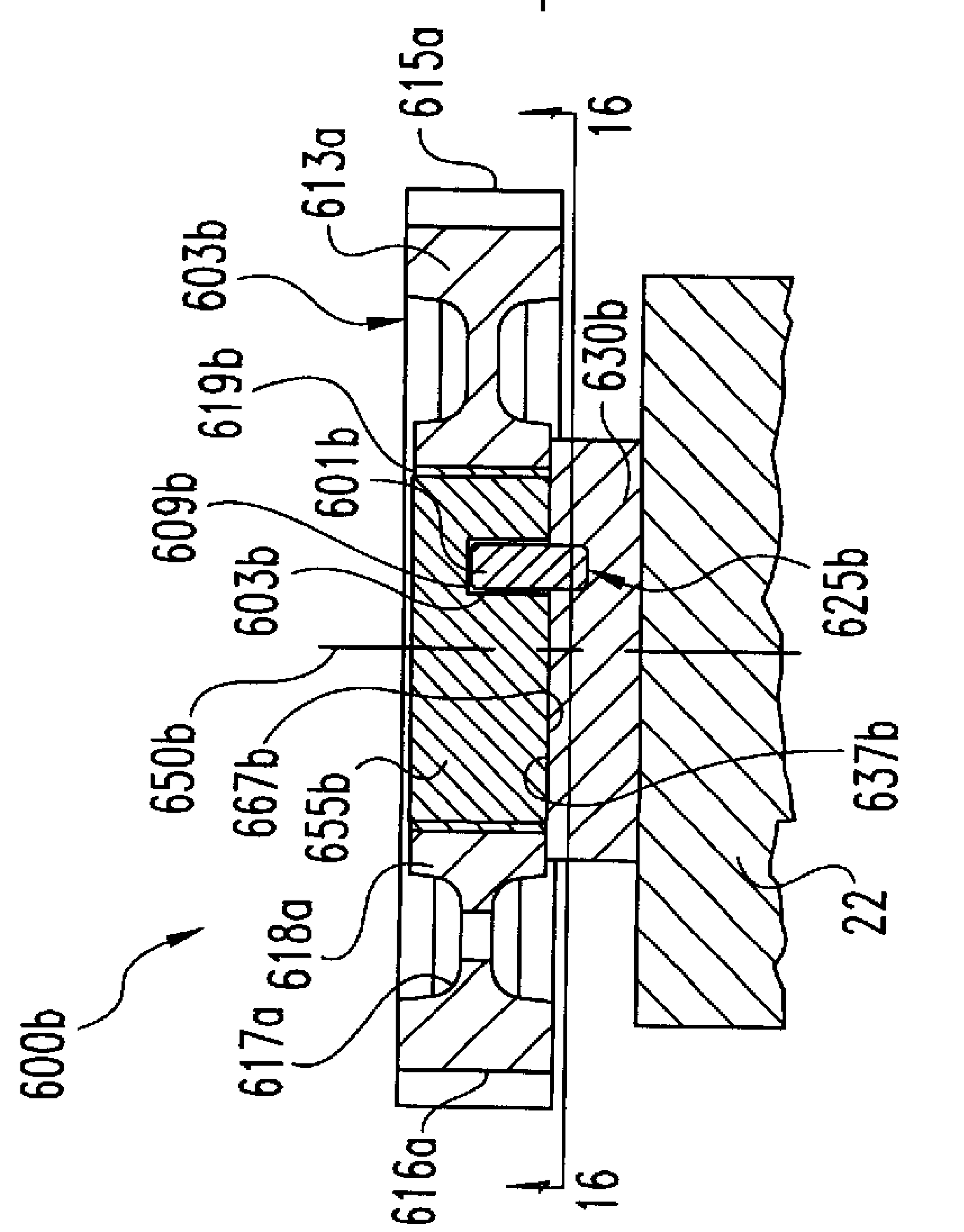
655b
614b
610b
50
612b
54

Fig. 16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630b
619b
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667b
655b
637b
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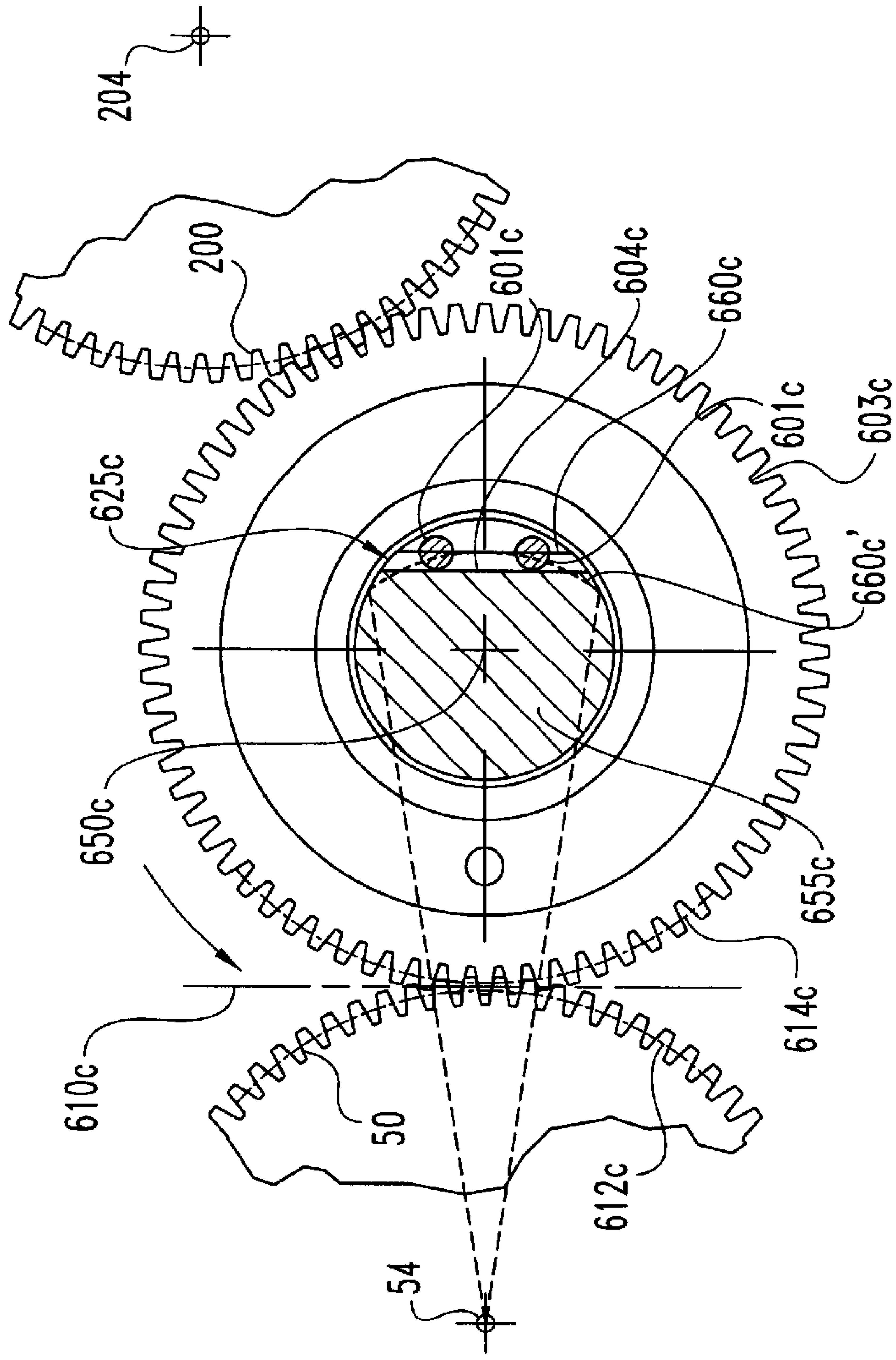
600b
618a
617a
616a
16

Fig. 15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604c
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603c
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660c'
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655c
614c
610c
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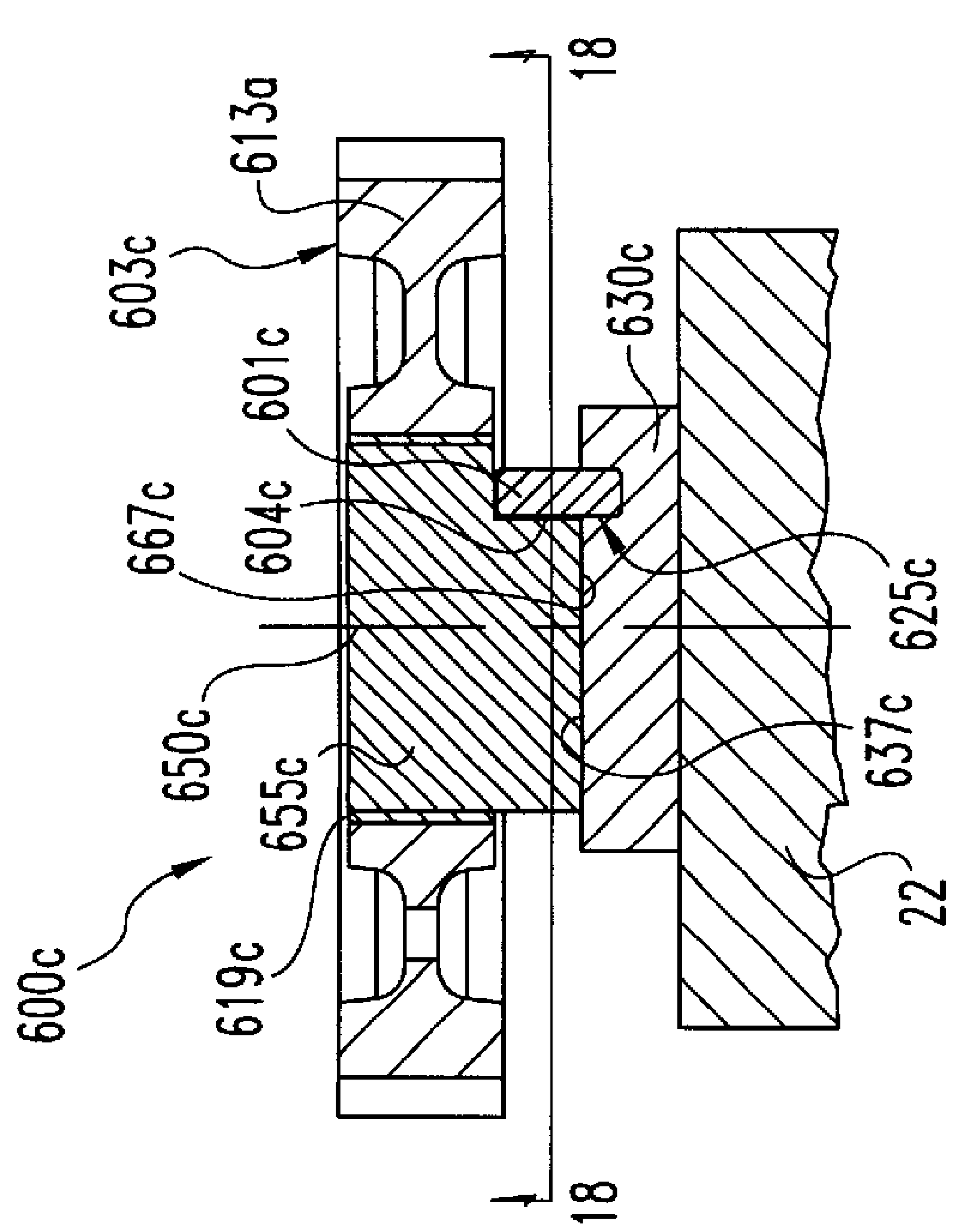
612c
54

Fig. 18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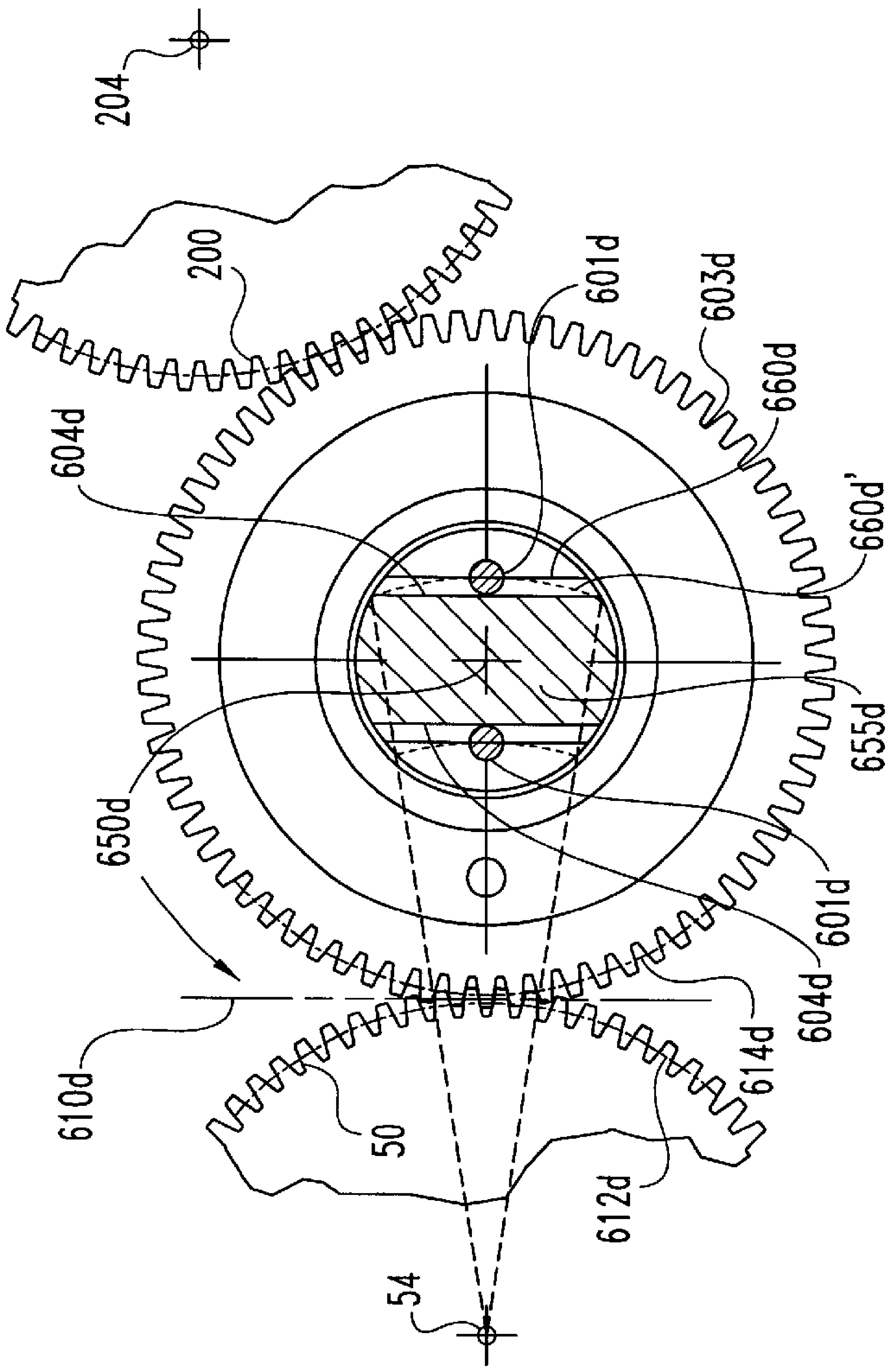
601c
604c
667c
625c
650c
637c
655c
600c
619c
22
18

Fig. 17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660d'
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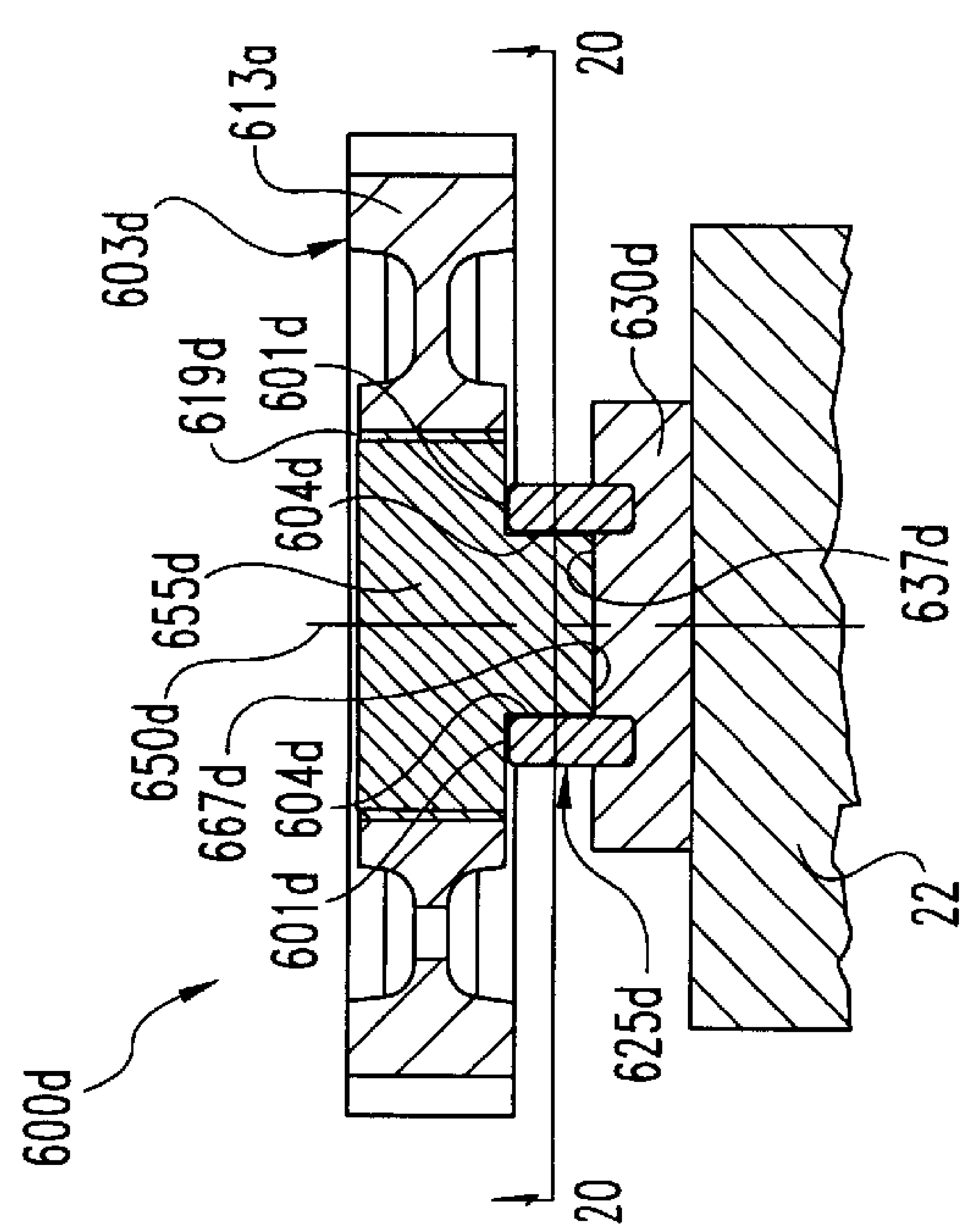
655d
650d
601d
604d
614d
610d
50
612d
54

Fig. 20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603d
619d
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650d
667d
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601d
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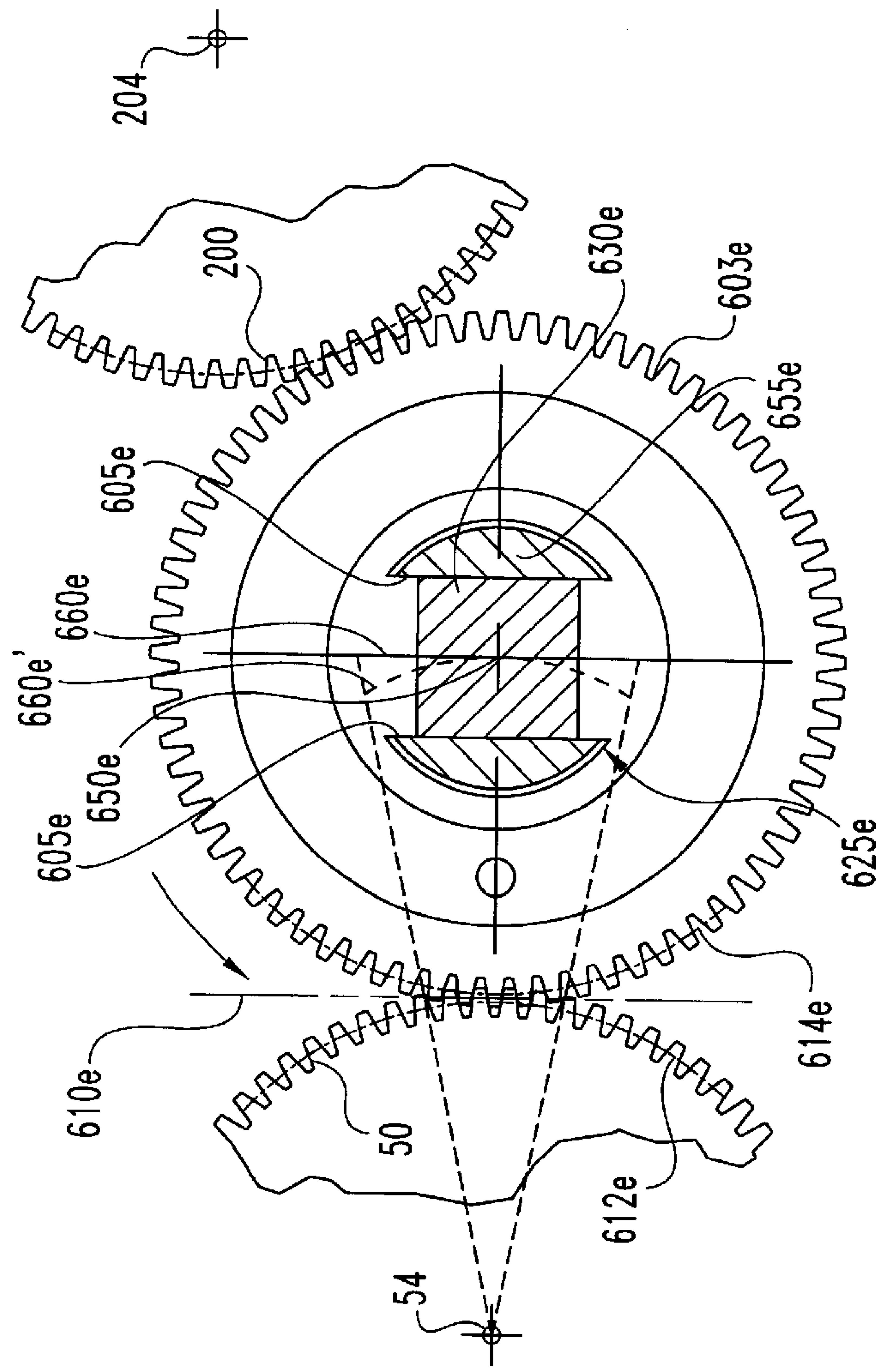
625d
600d
20

Fig. 19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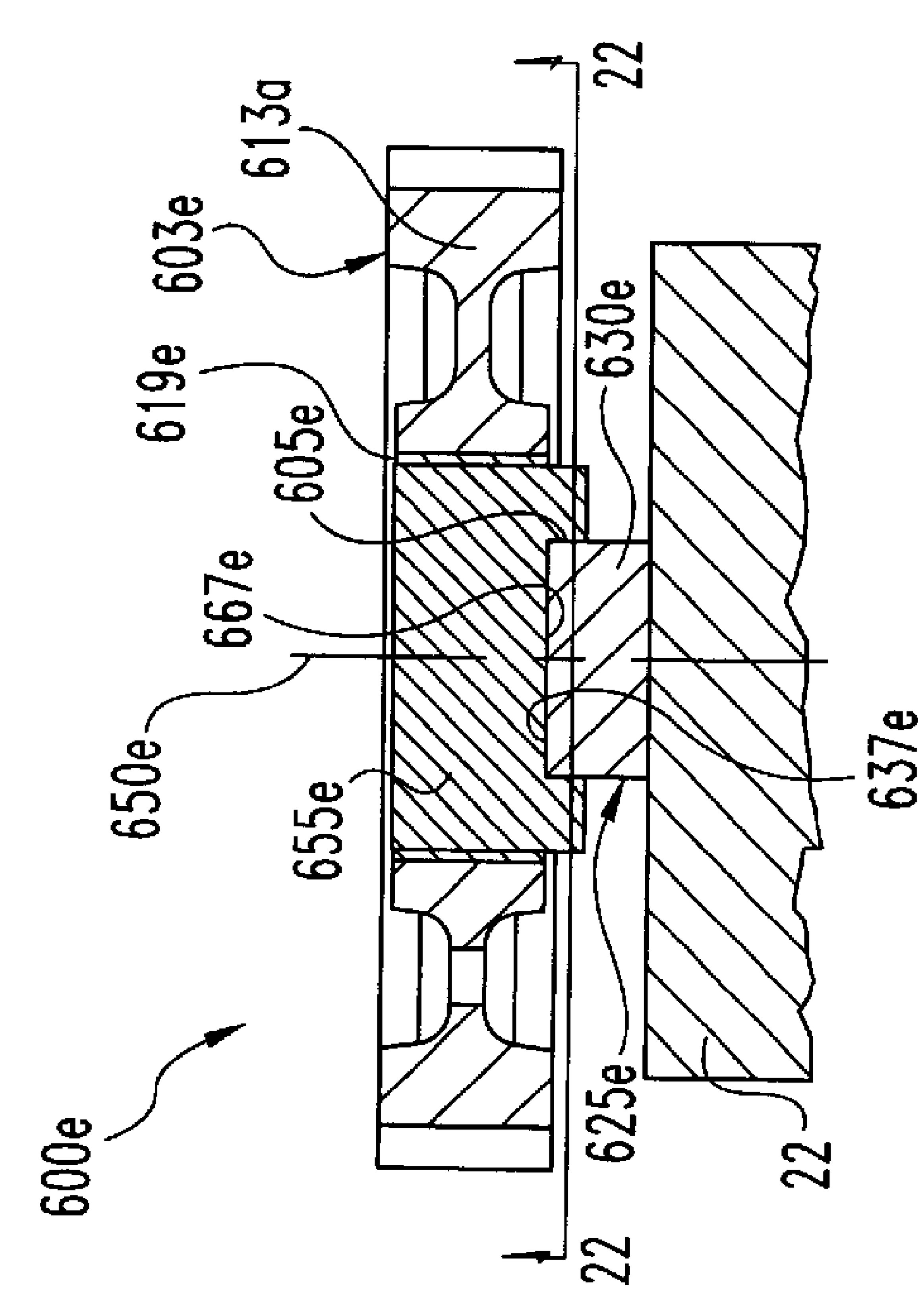
612e
54

Fig. 22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625e
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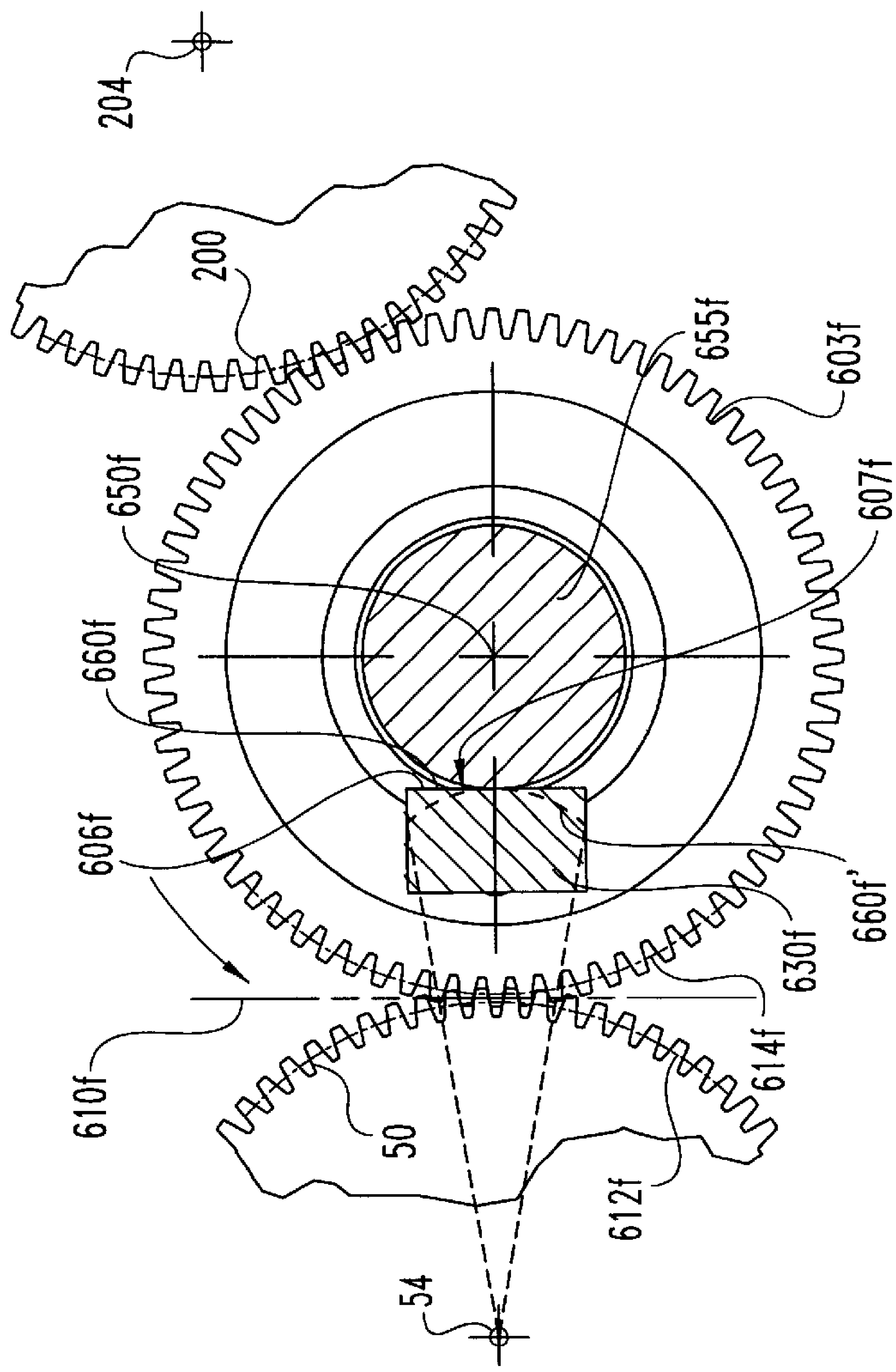
Fig. 21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612f
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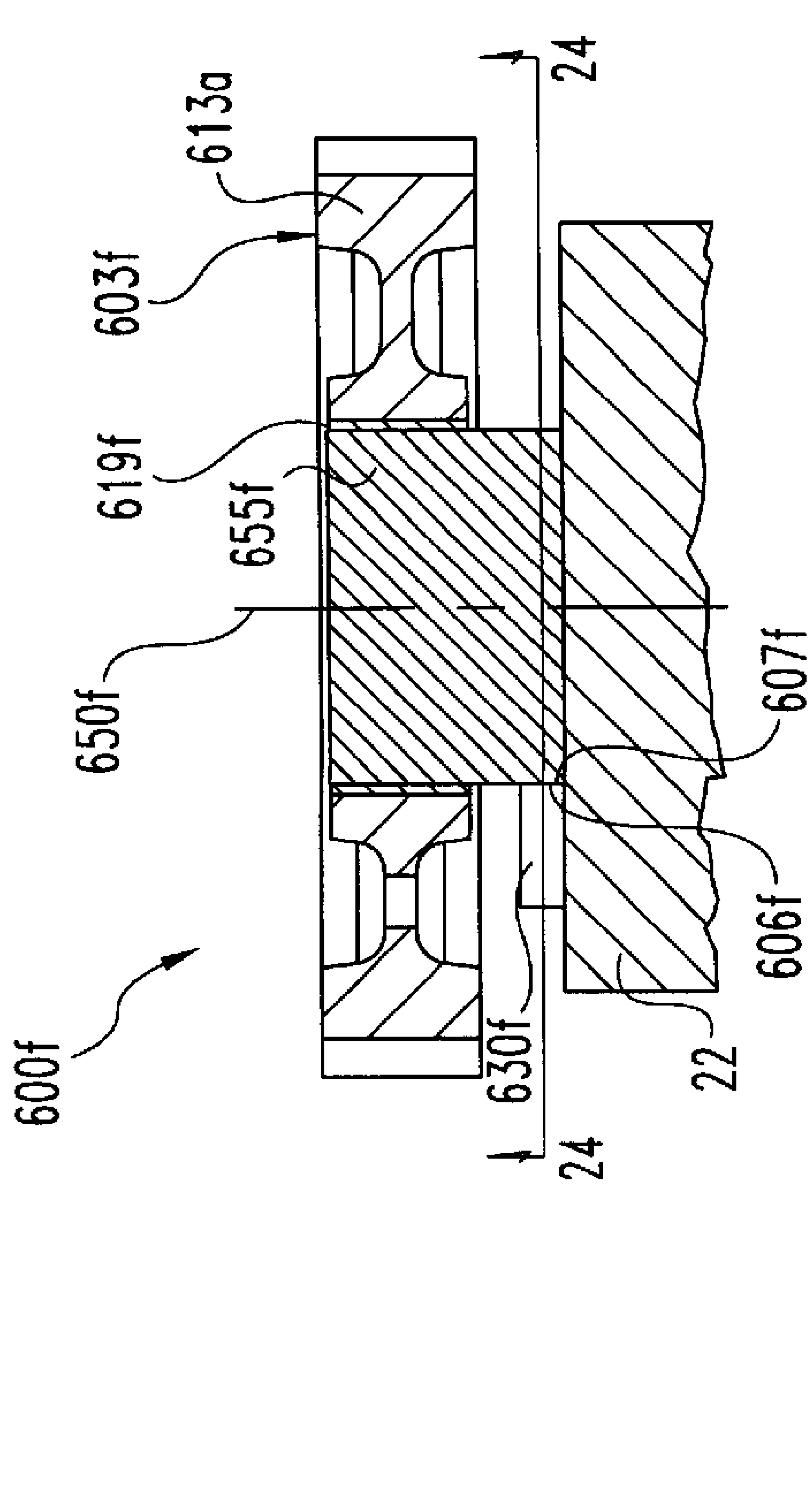
Fig. 24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606f
600f
630f
22
24

Fig. 23

APPARATUS AND METHOD FOR ADJUSTING A GEAR

REFERENCE TO RELATED APPLICATIONS

This application is a continuation of U.S. patent application Ser. No. 09/186,238 entitled APPARATUS AND METHOD FOR ADJUSTING A GEAR filed on Nov. 4, 1998, now U.S. Pat. No. 6,109,129, which is a continuation-in-part of U.S. patent application Ser. No. 08/853,341 entitled ANTI-LASH GEAR WITH ALIGNMENT DEVICE filed on May 8, 1997, now U.S. Pat. No. 5,870,928, and a continuation-in-part of U.S. patent application Ser. No. 08/853,013 entitled GEAR TRAIN ASSEMBLY INCLUDING SCISSOR GEAR filed on May 8, 1997, now U.S. Pat. No. 5,979,259. Additionally, this application is related to U.S. patent application Ser. No. 08/853,378 entitled ANTI-LASH GEAR ASSEMBLY filed on May 8, 1997, now U.S. Pat. No. 5,979,260.

BACKGROUND OF THE INVENTION

The present invention relates to gears, and more particularly, but not exclusively, relates to reduction of backlash in gear trains.

When the tooth of one gear mates with the gap of another gear, the gap typically provides more space than needed to accommodate the tooth. This excess space is sometimes called "lash" or "backlash." Backlash may vary with a number of factors including radial play in the gear bearings, gear shaft eccentricity, incorrect center-to-center spacing of the gears, and the gear-to-gear variation typical of many gear manufacturing processes.

The extra space associated with backlash usually leads to significant impact loading of the gear teeth. This loading often causes excessive noise and may result in other gear train problems. For example, backlash may accelerate gear wear. Backlash reduction is of particular concern for internal combustion engine applications—especially for gear trains used with diesel engines. U.S. Pat. No. 5,450,112 to Baker et al., U.S. Pat. No. 4,920,828 to Kameda et al., U.S. Pat. No. 4,700,582 to Bessette, and U.S. Pat. No. 3,523,003 to Hambric are cited as sources of background information concerning the application of gear trains to various engines.

One way to reduce backlash is through precision machining and mounting of the gears. However, this approach is usually expensive and still may not adequately address backlash that changes over time due to wear. Another approach to reduce backlash has been the introduction of one or more scissor gears into the gear train. Generally, scissor gears have teeth which adjust in size to occupy the space available between teeth of a mating gear. U.S. Patent No. 5,056,613 to Porter et al., U.S. Pat. No. 4,747,321 to Hannel, U.S. Pat. No. 4,739,670 to Tomita et al., U.S. Pat. No. 3,365,973 to Henden, and U.S. Pat. No. 2,607,238 English et al. are cited as examples of various types of scissor gears.

Backlash accommodation with a scissor gear is often limited when the scissor gear is meshed with two or more gears having different amounts of lash. Typically, the mating gear having the smallest amount of lash dictates the effective tooth size of the scissor gear; however, this size is generally inadequate to take-up the greater lash of the other mating gear or gears. One potential solution to this problem is to select mating gears which minimize the lash difference, but this "lash matching" approach is typically expensive and time-consuming. Consequently, a need remains for a gear train assembly which accommodates lash differences resulting from multiple gears meshing with a scissor gear.

One scissor gear configuration has two toothed wheels spring-biased to rotate relative to each other about a common center. For this configuration, paired gear teeth, one from each wheel, spread to occupy the available space between teeth in a mating gear. In some gear trains, loading of the tooth pairs by the mating gear becomes high enough to align each tooth pair in opposition to the spring bias. Typically, each member of the aligned pair is configured to proportionally bear this high load by being sized with the same nominal thickness. However, it has been found that random deviations from nominal are usually enough to cause one tooth or the other of each pair to bear a disproportionately high amount of the load until it has deformed enough to match the other tooth. This deformation process often subjects the gear teeth to reverse bending loads that more quickly wear-out the teeth compared to teeth subjected to unidirectional bending loads. Also, such deformation may cause greater tooth-to-tooth variation, resulting in poorer performance and a more noisy gear train. Therefore, a need exists for an anti-lash gear assembly which accommodates high loading without these drawbacks.

It has also been discovered that the knocking of heavy duty diesel engines, often attributed to combustion processes, results, at least in part, from high impact gear tooth noise. Typically, this noise is not sufficiently abated by conventional scissor gear configurations. Thus, a gear train is also in demand which addresses this type of noise.

SUMMARY OF THE INVENTION

The present invention relates to anti-lash gear assemblies and gear trains utilizing one or more anti-lash gear assemblies. Various aspects of the invention are novel, nonobvious, and provide various advantages. While the actual nature of the invention covered herein can only be determined with reference to the claims appended hereto, certain forms of the invention that are characteristic of the preferred embodiments disclosed herein are described briefly as follows.

One form of the present invention is a gear train having a first gear forming a first mesh with a second gear and a second mesh with a third scissor gear. A mounting position of the first gear relative to the second and third gears is selected to maintain backlash of both the first and second meshes at or below a maximum acceptable level.

In an another form, an idler gear forms a first mesh with a first scissor gear to establish an effective tooth size of the first scissor gear. After the first mesh is established, the mounting position of the idler gear is selected. This mounting position is determined as a function of the effective tooth size to control backlash of a second mesh formed between the idler gear and a second scissor gear.

In a further form, an idler gear forms a first mesh with a first gear and a second mesh with a second gear. A mounting position of the idler gear relative to the first and second gears is selected to maintain backlash of both the first and second meshes at or below a maximum acceptable level. This mounting position may be selected by using a mechanism that constrains movement of the idler gear along a desired path and the first gear, second gear, or both may be of a scissor type configuration.

In an additional form of the present invention, a gear train assembly for an engine is provided which includes a first scissor gear rotatably coupled to the engine, the first scissor gear having a first rotational center; a second scissor gear rotatably coupled to the engine, the second scissor gear having a second rotational center; and an idler gear rotatably coupled to the engine to form a first mesh with the first scissor gear and a second mesh with the second scissor gear. The idler gear has a third rotational center and is mounted by a positioning mechanism having a guide member slidingly engaging a generally linear adjustment path, the idler gear being selectively positionable along the path to maintain a distance between the first and third rotational centers within a predetermined range corresponding to a generally minimized backlash for the first mesh, and to provide a correspondingly adjustable separation distance range between the second and third rotational centers to generally match backlash of the second mesh to the minimized backlash of the first mesh.

In yet another form of the present invention, a method of assembling a gear train includes: (a) mounting a first gear which defines a first rotational center; (b) mounting a second gear which defines a second rotational center; and (c) mounting a third gear defining a third rotational center to establish a first mesh of a generally minimized backlash with the first gear and a second mesh with the second gear, the third gear being adjustable about a generally linear adjustment path, the linear adjustment path having a predetermined relationship to a tangent formed by the first mesh, whereby the third gear is selectively positionable along the adjustment path to minimize a backlash of the second mesh while maintaining the generally minimized backlash of the first mesh.

In still another form of the present invention, a gear train assembly for an engine includes a first gear rotatably coupled to the engine, the first gear having a first rotational center; a second gear rotatably coupled to the engine, the second gear having a second rotational center; and an idler gear rotatably coupled to the engine to form a first mesh with the first gear and a second mesh with the second gear. The idler gear defines a third rotational center and includes means for positioning the third gear along a generally linear adjustment path, the idler gear being selectively positionable along the path to maintain a distance between the first and third rotational centers within a predetermined range corresponding to a generally minimized backlash for the first mesh and to provide a correspondingly adjustable separation distance range between the second and third rotational centers to generally match backlash of the second mesh to the minimized backlash of said first mesh.

Other forms of the present invention include incorporating the various anti-lash gear assemblies of the present invention into a gear train and utilizing the adjustable positioning mechanisms of the present invention with an internal combustion engine.

It one object of the present invention to reduce noise emitted by engine gear trains.

Another object of the present invention is to provide an anti-lash gear assembly which reduces gear train noise emissions.

It is another object of the present invention to control load sharing between multiple gear wheels of a scissor gear assembly.

Yet another object is to provide a reliable technique for adjusting gear assemblies to reduce backlash.

Further objects, features, advantages, benefits and aspects of the present invention will become apparent from the drawings and description contained herein.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a front elevational view of an internal combustion engine system of one embodiment of the present invention.

FIGS. **2** and **3** are top plan views of components of an anti-lash gear assembly for the embodiment of FIG. **1**.

FIG. **4** is a top plan view of the components of FIGS. **2** and **3** incorporated into the anti-lash gear assembly in an unaligned configuration.

FIG. **5** is a perspective view of the anti-lash gear assembly of FIG. **4** in an aligned configuration.

FIG. **6** is an cross-sectional view of an idler gear and adjustable positioning mechanism along section lines **6—6** of FIG. **1**.

FIGS. **7**A and **7**B are schematic, front elevational views of the system of FIG. **1** at various stages of assembly.

FIGS. **8**A–**8**C are schematic, front elevational views depicting selected operational states of a portion of the system of FIG. **1**.

FIG. **9** is a graph illustrating various relationships concerning the operational states shown in FIGS. **8**A–**8**C.

FIG. **10** is an exploded perspective view of an anti-lash gear assembly of an alternative embodiment of the present invention.

FIG. **11**A is a top plan view of the anti-lash gear assembly of FIG. **10** in an unaligned configuration.

FIG. **11**B is a side elevational view of the anti-lash gear assembly of FIG. **11**A.

FIG. **12**A is a top plan view of the anti-lash gear assembly of FIG. **10** in an aligned configuration.

FIG. **12**B is a side elevational view of the anti-lash gear assembly of FIG. **12**A.

FIG. **13** is a section view of a second embodiment of an idler gear adjustment mechanism of the present invention.

FIG. **14** is a cross-sectional view of the gear adjustment mechanism of FIG. **13** taken through section line **14—14**.

FIG. **15** is a section view of a third embodiment of a gear adjustment mechanism of the present invention.

FIG. **16** is a cross-sectional view of the gear adjustment mechanism of FIG. **15** taken through section line **16—16**.

FIG. **17** is a section view of a fourth embodiment of a gear adjustment mechanism of the present invention.

FIG. **18** is a cross-sectional view of the gear adjustment mechanism of FIG. **17** taken through section line **18—18**.

FIG. **19** is a section view of a fifth embodiment of the gear adjustment mechanism of the present invention.

FIG. **20** is a cross-sectional view of the gear adjustment mechanism of FIG. **19** taken through section line **20—20**.

FIG. **21** is a section view of a sixth embodiment of the gear adjustment mechanism of the present invention.

FIG. **22** is a cross-sectional view of the gear adjustment mechanism of FIG. **21** taken through section line **22—22**.

FIG. **23** is a section view of a seventh embodiment of the gear adjustment mechanism of the present invention.

FIG. **24** is a cross-sectional view of the gear adjustment mechanism of FIG. **23** taken through section line **24—24**.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

For the purposes of promoting an understanding of the principles of the invention, reference will now be made to the embodiments illustrated in the drawings and specific language will be used to describe the same. It will nevertheless be understood that no limitation of the scope of the invention is thereby intended. Any alterations and further modifications in the described embodiments, and any further applications of the principles of the invention as described herein are contemplated as would normally occur to one skilled in the art to which the invention relates.

FIG. **1** depicts internal combustion engine system **20** of the present invention. System **20** includes engine block **22** with a crankshaft **24** shown in phantom. Engine system **20** also includes head assembly **30** connected to block **22**. Head assembly **30** includes fuel injector camshaft **32** shown in phantom and valve camshaft **34** shown in phantom. In one embodiment, block **22** and head assembly **30** are configured as a heavy duty, in-line six cylinder diesel engine. The present invention is also applicable to other types of engines as would occur to one skilled in the art.

System **20** includes timing gear train **40**. Gear train **40** includes drive gear **42** connected to crankshaft **24**. Crankshaft **24** and drive gear **42** have rotational center **44** at the intersection of the crosshairs designated by reference numeral **44**. For the figures referenced herein, centers of rotation are depicted by a broken line segment indicative of the corresponding rotational axis when the rotational axis is not perpendicular to the view plane and by crosshairs when the rotational axis is perpendicular to the view plane. Gear **42** rotates with crankshaft **24** during operation of engine system **20** about center **44** to drive the remaining gears of gear train **40**.

Gear **42** has teeth **46** which form mesh **48** with lower idler anti-lash gear **50**. Gear **50** rotates about shaft **53** having rotational center **54**. Shaft **53** is mounted to block **22** by fasteners **55**. Bearing **56** provides a rotational bearing relationship between anti-lash gear assembly **58** of gear **50** and shaft **53**.

FIGS. **2–5** provide additional details concerning the structure and operation of anti-lash gear assembly **58** of gear **50**. Referring to FIG. **2**, various details of gear wheel **60** prior to incorporation into gear assembly **58** are shown. Gear wheel **60** includes a hub **63**. Web **64** defines seven circumferentially spaced apart apertures **65**. Furthermore, for each aperture **65**, web **64** defines a fingered edge **65***a* at one end opposing edge **65***b* at another end. Aperture **65** and edges **65***a*, **65***b* are generally evenly spaced along the circumference of an imaginary circle about center **54**. Gear wheel **60** includes a number of circumferentially spaced-apart gear teeth **66** defined by rim **67**. Rim **67** is integrally connected to hub **63** by web **64**. Adjacent members of gear teeth **66** are generally evenly spaced-apart from one another by gaps **68**. Only a few of teeth **66** and gaps **68** are designated to preserve clarity. Each member of gear teeth **66** is generally sized and shaped the same as the others. Similarly, each gap **68** generally has the same size and shape.

Referring to FIG. **3**, gear wheel **70** of anti-lash assembly **58** is illustrated. Gear wheel **70** includes hub **73** which is configured to form a rotary bearing relationship with shaft **53** via bearing **56** (see FIG. **1**). Hub **63** of gear wheel **60** engages hub **73**. The interface between hubs **63** and **73** is adapted to permit rotation of gear wheels **60** and **70** relative to each other. Gear wheel **70** also includes web **74**. Tabs **74***a* project from web **74** in a direction generally perpendicular to the view plane of FIG. **3** and have one side connected to rim **77** to define corresponding recesses **75**. At least one tab **74***a* defines threaded bore **79** therethrough. Bore **79** has a longitudinal axis generally parallel to the view plane of FIG. **3**. Web **74** also defines lightening holes **75***a* each corresponding to one of recesses **75**. Tabs **74***a* and recesses **75** are generally evenly spaced along the circumference of an imaginary circle about center **54**.

Wheel **70** includes a number of gear teeth **76** defined by rim **77**. Rim **77** is integrally connected to hub **73** by web **74**. Adjacent members of gear teeth **76** are generally evenly spaced-apart from one another by gaps **78**. Only a few of teeth **76** and gaps **78** are designated to preserve clarity. Each member of gear teeth **76** is generally sized and shaped the same as the others. Similarly each gap **78** generally has the same shape and size. Preferably, the number of teeth **76** of wheel **70** is the same as the number of teeth **66** of wheel **60**.

FIG. **4** defines anti-lash gear assembly **58** in an unaligned configuration commonly encountered prior to preparation for installation in gear train **40**. In this configuration, wheels **60** and **70** loosely engage each other so that each aperture **65** of wheel **60** generally overlaps a corresponding recess **75** of wheel **70** to define a number of pockets **80**. A number of coil springs **81** are provided each having end **82** opposite end **84**. Each spring **81** is positioned in a corresponding one of pockets **80** with end **82** engaging a corresponding tab **74***a* and end **84** aligning with a corresponding edge **65***a*. However, ends **84** do not typically engage edges **65***a* in this configuration.

Assembly **58** also includes adjustment bolt **90** having threaded stem **92** opposing head **94**. Stem **92** is shown fully threaded into bore **79** in FIG. **4** with head **94** in contact with corresponding tab **74***a*. By convention, teeth **66** and **76** are in an "unaligned" position such that teeth **66** overlap gaps **78** defined between teeth **76**, and teeth **76** overlap gaps **68** defined between teeth **66**. Hub **73** of wheel **70** forms a rotary bearing relationship with hub **63** of wheel **60** so that wheels **60** and **70** are permitted to rotate relative to one another. Head **94** defines contact surface **95** configured to bear against adjacent edge **65***b* of wheel **60** when wheel **60** is rotated counter-clockwise relative to wheel **70**. When wheel **60** is rotated in the clockwise direction relative to wheel **70**, spring ends **84** eventually engage corresponding edges **65***a*. Preferably, each edge **65***a* defines a finger sized to fit inside the coil of each spring **81** to facilitate proper alignment with wheel **60**. When rotated in the clockwise direction with sufficient force, springs **81** are compressed between corresponding edges **65***a* and tabs **74***a*, as illustrated in FIG. **5**.

FIG. **5** depicts an "aligned" position of gear wheel **60** and **70** reflecting a configuration suitable for installation in gear train **40**. When aligned, teeth **76** and **66** are approximately centered over one another as depicted in FIG. **5**. Springs **81** are also in a highly compressed condition between edges **65***a* and tabs **74***a* to provide a correspondingly high spring force. Adjusting assembly **58** from the configuration of FIG. **4** to the configuration of FIG. **5** is provided by unthreading bolt **90** so that head **94** moves away from bore **79** along stem axis S. As this unthreading continues, surface **95** bears against adjacent edge **65***b* and springs **81** are compressed between adjacent aligned tabs **74***a* and edges **65***a*.

Unthreading of bolt **90** spreads apart the associated tab **74***a* and edge **65***b* to rotate wheels **60** and **70** rotate relative to one another and move teeth **66** and **76** past each other. A given tooth of wheel **66** may move into and out of registration with several teeth **76** before reaching the highly biased configuration of FIG. **5** from the unbiased configuration of FIG. **4**.

FIG. **5** also depicts face **66***a* of each tooth **66** of wheel **60** a few of which are depicted. Each tooth **76** of wheel **70** similarly has a face **76***a*, a few of which are depicted. Width **W60** represents the width of a typical face **66***a*. Similarly, width **W70** represents the width of a typical face **76***a*. Preferably, width **W60** is less than width **W70**. More preferably, width **W70** is at least about 50% greater than width **W60**. Most preferably, width **W70** is at least about twice width **W60**.

Referring collectively to FIGS. **4** and **5**, anti-lash gear wheel assembly **58** is constructed by providing wheel **70** and mounting one of springs **81** to align with bore **79**. Bolt **90** is threaded into bore **79** so that head **94** contacts the associated tab **74***a*. The remaining springs **81** are placed in recesses **75** of wheel **70**. Wheel **60** is placed over wheel **70** to define corresponding pockets **80** generally evenly spaced along imaginary circle **86** (shown in phantom in FIG. **4**). Edges **65***a* align with ends **84** of corresponding springs **81**.

Prior to mounting assembly **58** on shaft **53**, it is preferred that teeth **66** and **76** be aligned. To provide this alignment, bolt **90** is partially unthreaded from bore **79** so that head **94** contacts adjacent edge **65***b* of wheel **60** and correspondingly compresses springs **81**. In response, teeth **66**, **76** move past one another. Unthreading of bolt **90** continues this motion until the aligned position of FIG. **5** is generally reached. As a result, wheel **60** is separated from wheel **70** along stem axis S by distance D as illustrated in FIG. **5**. Notably, a portion of stem **92** of bolt **90** remains threaded in bore **79** in both the unaligned position of FIG. **4** and in the aligned position of FIG. **5**. In other embodiments, more than one or all of tabs **74***a* may be adapted to define a bore **79** suitable for engagement by bolt **90**. Similarly, multiple bolts **90** may be employed with embodiments having multiple bores **79**.

Once teeth **66** and **76** are aligned in the configuration of FIG. **5**, assembly **58** is mounted to shaft **53** via bearing **56**. When so mounted, the aligned teeth **66**, **76** form mesh **48** with teeth **46** of drive gear **42**. However, mesh **48** typically has a significant amount of lash when teeth **66**, **76** are forcibly aligned by the extension of bolt **90**. To take-up this lash with gear **50**, wheels **60** and **70** are preferably permitted to rotate relative to one another under the influence of the bias provided by compressed springs **81**. Threading bolt **90** back into bore **79** once assembly **58** is mounted to form mesh **48** with drive gear **42** permits this rotation. As a result, the spring bias offsets teeth **66** and **76** from one another to generally occupy the entire space between adjacent teeth **46** participating in mesh **48**. Notably, mesh **48** does not permit teeth **66**, **76** to return to the unloaded position of the FIG. **4** configuration.

Each pair of initially aligned teeth **66**, **76** operate collectively as a composite tooth with a variable effective size or "thickness" dependent upon the space between mating teeth **46**. By varying in thickness, these composite teeth may reduce, or even effectively eliminate, backlash in mesh **48**. To conclude installation of assembly **58**, bolt **90** should be tightened down so that head **94** bears against the associated tab **74***a*. Bolt **90** is preferably carried by wheel **70** throughout the adjustment process and utilization of assembly **58** as part of gear **50**.

Preferably, wheel **60** and **70** are machined from a metallic material suitable for long-term use in a diesel engine timing gear train. It is also preferred that bolt **90** and springs **81** be selected from compatible materials suitable for long term use in a diesel engine environment. Nonetheless, in other embodiments, different materials may be used as would occur to one skilled in the art.

Although gear **50** is illustrated in FIG. **1** as an idler gear, in other configurations it may be configured as a driving gear, a driven gear, or otherwise adapted or modified as would occur to one skilled in the art. In all these forms, gear **50** may be considered to be a novel type of "scissor gear."

Referring back to FIG. **1**, gear **50** participates in gear train **40** to form mesh **96** with idler gear **100**. Idler gear **100** rotates about rotational center **104** and defines circumferential teeth **106** spaced apart by gaps **108** to form mesh **96** with gear **50**.

Referring additionally to FIG. **6**, further details concerning idler gear **100** are provided. Idler gear **100** includes rim **107** defining teeth **106** integrally connected to web **114**. Web **114** defines lightening holes **116**. Web **114** is also integrally connected to hub **118** which, as shown in the cross-sectional view of FIG. **6**, has slightly less thickness along the rotational axis corresponding to center **104** than rim **107**. Cylindrical bushing **119** provides a rotational bearing surface between shaft **103** and hub **118**. Shaft **103** defines four passages **105** used to mount idler gear **100** to block **22**.

Mounting of idler gear **100** is provided by adjustable positioning mechanism **120**. Mechanism **120** includes a mounting plate **130** which is positioned between shaft **103** of idler gear **100** and block **22**. Notably, plate **130** is configured to provide clearance with hub **118** of idler gear **100** so that idler gear **100** may freely rotate about shaft **103**.

Idler gear **100** and mounting plate **130** are positioned between block **22** and retaining plate **140**. Retaining plate **140** includes mounting holes **145** which are generally aligned with mounting passages **105** of shaft **103**, mounting passages **135** of plate **130**, and threaded bores **25** of block **22**. Notably passages **105** have a larger dimension along an axis perpendicular to the rotational axis of gear **100** than passages **135**, holes **145**, and bores **25**. Idler gear **100** is secured between plates **130** and **140** by inserting cap screw fasteners **150** through holes **145**, passages **105**, and passages **135** and threading the end of threaded stems **152** into bores **25**. Fasteners **150** each have head **154** opposing threaded stem **152**. Head **154** is sized to contact retaining plate **140** when stems **152** are fully threaded into bores **25** to clamp plate **140** against shaft **153** and to clamp shaft **153** against plate **130**.

In operation, mechanism **120** is configured to position idler gear **100** relative to a planar region that is preferably parallel to the view plane of FIG. **1** and perpendicular to the view plane of FIG. **6**. Within this region, gear **100** may be positioned with two degrees of freedom as symbolized by the X and Y directional arrows of FIG. **1**.

To mount idler gear **100**, mounting plate **130** is first secured to block **22** using fasteners (not shown) in a conventional manner so that passages **135** align with bores **25**. Once plate **130** is secured to block **122**, idler gear **100** is located on plate **130** so that passages **105** overlap passages **135**. Next, retaining plate **140** is placed over shaft **103** to locate holes **145** over corresponding passages **105** and **135**, and bores **25**. Fasteners **150** are then each placed through an aligned hole **145**, passage **105**, and passage **135** and loosely threaded into a corresponding bore **25**. Preferably, fasteners **150** are initially threaded into bores **25** an amount sufficient to contact plate **140** and yieldingly hold idler gear **100** in position. In this configuration, the position of idler gear **100** relative to the planar region symbolized by the X and Y directional arrows may be selected within the range permitted by the clearance of fasteners **150** in passages **105**. Once an X-Y position is selected, fasteners **150** are tightened down to secure idler gear **100** and mechanism **120**.

Teeth **106** of idler gear **100** form mesh **196** with anti-lash gear **200**. Gear **200** is mounted to fuel injector camshaft **32** of head assembly **30** and is configured to rotate about rotational center **204**. Gear **200** is preferably configured similar to gear **50** having composite gear tooth pairs represented by reference numeral **266**. Furthermore, springs **281** of gear **200** are shown configured in a manner similar to springs **81** of gear **50**, although fewer in number (three being shown). Likewise an installation adjustment bolt **290** is shown. This adjustment bolt may function for installation purposes similar to bolt **90** of gear **50**. Gear **50**, gear **200**, or both may utilize belleville washers to provide a spring bias either with or without coil springs.

Gear **200** forms mesh **296** with mating gear **300**. Mating gear **300** is attached to valve camshaft **34** to rotate about rotational center **304**. Gear **300** defines teeth **306** which interface with tooth pairs **266** of gear **200** to form mesh **296**.

In operation, drive gear **42** rotates with crankshaft **24** to turn gear **50**. In response, gear **50** turns idler gear **100** via mesh **96**. Idler gear **100** drives gear **200** via mesh **196** to regulate timing of fuel injectors (not shown) for engine system **20** by rotating fuel injector camshaft **32**. Furthermore, gear **200** drives mating gear **300** via mesh **296** to rotate valve camshaft **34** therewith to time engine valves (not shown) for head assembly **30**. Thus, gear train **40** turns camshafts **32** and **34** of head assembly **30** in response to rotation of crankshaft **24** to control timing of engine system **20**.

In other embodiments, different quantities and arrangements of gears in gear train **40** may be utilized as would occur to one skilled in the art. In one alternative embodiment, a conventional scissor gear may be used in place of gear **50**, gear **200**, or both. In still other embodiments an idler gear with an adjustable positioning mechanism may not be required.

In one embodiment of gear train **40**, the number of teeth **46** is about **48** for drive gear **42**; the number of teeth **66, 76** is about **70** for gear wheels **60, 70**, respectively; the number of teeth **106** for adjustable idler gear **100** is about **64**; the number of composite teeth **266** for gear **200** is about **76** and the number of teeth **306** is about **76** for gear **300**. Furthermore, for this configuration, gears **42, 50, 100, 200, 300** are of a spur gear configuration, are made from metallic materials suitable for long term use with internal combustion engines, and have generally parallel rotational axes which perpendicularly intersect the view plane of FIG. **1**.

Having described selected structural and operational features of system **20**, certain aspects concerning the assembly of system **20** are next described in connection with the schematic representations of FIGS. **7**A and **7**B. In FIGS. **7**A and **7**B, reference numerals schematically depict structure identified by like reference numerals in FIGS. **1–6**; however, gear meshes have been enlarged to emphasize selected features of the present invention. FIG. **7**A illustrates an intermediate assembly stage of drive train **40**. In this stage, drive gear **42** has been previously mounted to rotate about center **44** in the direction indicated by arrow **R1**. Similarly, mating gear **300** has been mounted to rotate about center **304** in the direction indicated by arrow **R5**.

After gears **42** and **300** have been mounted, gears **50** and **200** are mounted to form mesh **48** between gears **42** and **50**, and mesh **296** between gears **200** and **300**. The formation of meshes **48, 296** determines the effective composite tooth size of corresponding pairs of teeth for gears **50** and **200** as they occupy gaps between teeth **46** and **306** of gears **42** and **300**, respectively. For gear **50**, teeth **76** of wheel **70** are represented by dashed lines, and teeth **66** of wheel **60** are represented by solid lines for illustrative purposes. The effective circular thickness T**50** of one composite tooth pair of gear **50** is also shown. This composite circular thickness is determined along a pitch circle of gear **50** for mesh **48**. Notably, in the absence of idler gear **100**, thickness T**50** is defined by the mating gap of teeth **46** of gear **42**.

For mesh **296**, gear **200** forms composite teeth pairs **266**. Each pair **266** has a member represented by a dashed line and a member represented by a solid line to enhance clarity. The effective circular tooth thickness of one composite tooth pair **266** is shown as circular thickness T**200** relative to a pitch circle for gear **200**.

Arrows **R4, R5** indicate the rotational direction in which gears **200, 300** are driven, respectively. Also indicated are mounting bores **25** of engine block **22** for reference.

Having defined the composite circular thicknesses T**50** and T**200**, idler gear **100** is installed to form mesh **96** with gear **50** and mesh **196** with gear **200** as depicted in FIG. **7**B. The tooth thicknesses T**50** and T**200** are typically different corresponding to a difference in the amount of backlash in meshes **48** and **296**. By using mechanism **120** to adjust the X-Y position of rotational center **104** relative to fixed rotational centers **54** and **204**, idler gear **100** may be located to optimally mesh with the pre-defined tooth sizes of gears **50** and **200** despite any lash difference. Fasteners **150** of mechanism **120** are illustrated in FIG. **7**B for reference.

The positional adjustment of idler gear **100** relative to the other gears results in significant control over the amount of backlash in meshes **96** and **196**. When the backlash difference resulting from different T**50** and T**200** widths is within a certain range, backlash may be reduced, or even effectively eliminated, through proper placement of idler gear **100** along a planar region perpendicular to the rotational axes of the meshing gears.

Notably, while the preferred embodiment presents two meshes **96, 196** with idler gear **100**, in other embodiments this assembly method may be practiced to control backlash for a different quantity of meshing gears. For example, this assembly technique finds application in gear trains having only three gears oriented similar to gears **42, 50,** and **100**.

Referring to FIGS. **8**A–**8**C, selected operational states of gears **42, 50,** and **100** are schematically depicted with reference numerals representing structure designated by like numerals in FIGS. **1–6**; however, fewer and larger teeth are schematically illustrated in these figures to emphasize various features. Referring to FIG. **8**A, gears **42, 50, 100** are in a static (motionless) state relative to each other. Referring to mesh **48**, imaginary pitch circles C**1**, C**2**, C**3** are represented by dashed lines for gears **42, 50, 100**, respectively. The circular thickness T**50***a* of a pair of gear teeth **76, 66** of gear **50** is shown as an arc along the companion pitch circle C**2**. Arrows DF**1** represent the forces counter-acting the bias of gear **50** for the depicted static condition in FIG. **8**A. The static reaction forces of gear **100** are shown by arrows RF**1**. Also depicted is the circular thickness T**60** of a selected tooth **66**, and the circular thickness T**70** of a selected tooth **76**. It is preferred that circular thickness T**60** be nominally less than circular thickness T**70** for each tooth **60, 70,** respectively. In one preferred embodiment, T**60** is at least about two thousands (0.002) of an inch less than T**70**. More preferably, this difference is at least about four thousands (0.004) of an inch. Most preferably, this difference is in a range of about two to six thousands (0.002–0.006) of an inch.

In FIG. **8**B, drive gear **42** is rotating in the direction indicated by arrow R**1** to provide a resultant drive force represented by arrow DF**2**. In response, gear **50** is rotating in the direction indicated by arrow R**2** and gear **100** is rotating in the direction indicated by arrow R**3**. The resultant reaction force presented by gear **100** is represented by arrow RF**2**. The resultant forces DF**2** and RF**2** are of sufficient intensity to partially overcome the spring bias, causing compression of springs **81** of gear **50**. As a result, the circular thickness T**50***b* of the composite pairs of teeth of gear **50** decreases relative to thickness T**50***a* (T**50***b* is less than T**50***a*). As the magnitude of the force transmitted from drive gear **42** increases, gear teeth **66**, **76** continue to approach alignment.

In FIG. **8**C, the resultant driving force DF**3** of gear **42** and reaction force RF**3** of gear **100** compresses springs **81** by an amount sufficient to align gear teeth **66** and **76**. When so aligned, composite thickness T**50***c* results. T**50***c* is less than both T**50***a* and T**50***b*, and is generally equal to the circular thickness T**70** of teeth **76**. Springs **81** are generally fully compressed in the FIG. **8**C configuration; storing energy generally equivalent in amount to springs **81** in the configuration of FIG. **5**.

The smaller circular thickness of teeth **66** compared to teeth **76** (T**60**<T**70**) prevents loading of teeth **66** beyond the load provided by the compressed springs of FIG. **8**C. In contrast, teeth **76** bear any load in excess of the spring load. Limiting the load on teeth **66** to the spring bias generally reduces reverse bending loads commonly resulting from random dimensional differences of tooth pairs having each member nominally sized to the same circular thickness. Preferably, the wider tooth face W**70** of each tooth **76** is selected to bear the higher driving loads in excess of the spring bias; however, the total width increase (**W60**+**W70**) for gear is typically less than the width increase required to withstand reverse bending loads by a scissor gear that has the same nominal circular thickness for all teeth.

FIG. **9** graphically represents the typical effect of reduced circular thickness T**60** compared to circular thickness T**70** with load lines **400**. The dashed line **400** represents gear wheel **60** and the solid line **400** represents gear wheel **70**. Horizontal segments **410** correspond to the pre-loaded bias of gear **50** under the static conditions of FIG. **8**A. Sloped segments **420** correspond to the loading of teeth **66**, **76** between the static condition of FIG. **8**A and the aligned position of FIG. **8**C. FIG. **8**B represents one point along segments **420**. Once loading compresses springs **81** to align teeth as illustrated in FIG. **8**C, the loading on teeth **66** of gear wheel **60** flattens to the maximum load of springs **81** as indicated by segment **430**. At the same time, the thicker face W**70** of teeth **76** bears the high intensity loading as indicated by sloped segment **440**. By allowing wheel **70** to handle the high loads and limiting loading of wheel **60** with the circular thickness differential (**T70**–**T60**), reverse bending loads are typically reduced.

It has been found that much of the unpleasant noise, such as the "hammering" sounds associated with heavy-duty diesel engines, is due to high impact noise from gear trains associated with those engines. An unexpectedly dramatic change in sound quality is experienced, typically including a reduction in overall noise intensity, when a relatively high bias torque is provided by a scissor gear participating in the gear train. As used herein, "bias torque" is the magnitude of the torque provided by a spring-biased scissor gear assembly. The bias torque is determined as the magnitude of the cross product of the vectors corresponding to a radial distance from the rotational center of the gear to the teeth and the force acting tangential to a circle corresponding to the radius. Typically, the bias torque varies as a function of the amount of loading of the scissor gear bias. Preferably, the bias torque is at a maximum when the gear teeth are generally aligned in opposition to the bias. For the aligned configuration of teeth **66**, **76** in FIG. **5**, a radial vector T and a force vector F are illustrated which may be used to determine bias torque for assembly **58**.

It has been found that a maximum bias torque of at least 100 foot-pounds (ft-lbs) provides improved gear train noise character and intensity. More preferably, a maximum bias torque of at least about 200 ft-lbs is provided. Most preferably, a maximum bias torque of at least about 500 ft-lbs is provided. In one most preferred embodiment, gear **50** is configured with a maximum bias torque of about 700 ft-lbs, and gear **200** is configured with a maximum bias torque of about 200 ft-lbs. In many instances, the bias torque of the present invention obviates the need to use expensive enclosures and panels to mute unpleasant noise.

FIG. **10** provides an exploded perspective view of anti-lash gear assembly **558** about rotational center **554** of an alternative embodiment of the present invention. Assembly **558** includes gear wheel **560** with splines **561** defined by inner cylindrical surface **564** of hub **563**. Hub **563** defines opening **563***a* therethrough. Splines **561** are of the helical type oriented about center **554** and inclined relative to the rotational axis of wheel **560**. Hub **563** is integrally connected to web **564**. A number of circumferentially disposed teeth **566** are defined by rim **567** which is also integrally connected to web **564**. Teeth **566** are generally evenly spaced apart from each other about center **554** and each have generally the same size and shape. Between adjacent teeth **566** are gaps **568** which are also generally evenly spaced apart from one another and have generally the same shape and size. Web **564** of wheel **560** defines two opposing apertures **569** therethrough.

Assembly **558** also includes wheel **570**. Wheel **570** includes splines **571** defined by cylindrical outer surface **572** of hub **573**. Splines **561** are of the helical type oriented about center **554** and inclined relative to the rotational axis of wheel **570**. Splines **571** are inclined in generally the same manner as splines **561** to mate therewith. Hub **573** is configured to fit within opening **563***a* of hub **563** to mate splines **561** and **571**. Hub **573** defines opening **573***a* surrounded by inner cylindrical surface **574** for establishing a rotational bearing relationship with a mounting shaft. Wheel **570** also includes web **574** integrally connected to hub **573**. Teeth **576** are defined by rim **577** which is integrally connected to web **574**. Teeth **576** are generally evenly spaced apart from one another about rotational center **554** and each have generally the same size and shape. Teeth **576** define gaps **578** therebetween. Gaps **578** are generally evenly spaced apart from one another and each have generally the same size and shape. Collectively, hub **573**, web **574**, and rim **571** define a cylindrical recess **575**. Web **564** defines two opposing threaded recesses **579** each corresponding to one of apertures **569**.

Coil springs **580** are each placed in recess **575** and are generally evenly spaced apart from one another about center **554** between hub **573** and rim **577**. Adjustment devices **590***a*, **590***b* are included which each have adjustment bolt **590** with threaded stem **592**. Stem **592** has end **593** opposing head **594**. Devices **590***a*, **590***b* each include washer **596** configured for passage of stem **592** therethrough. In contrast, head **594** is sized so that it will not pass through washer **596**. Also, the outer diameter of washer **596** is dimensioned so that it will not pass through aperture **569**. Aperture **569** is sized to provide ample clearance for stem **592**, permitting selective positioning of stem **592** therein. Threaded recesses **579** are each configured for engagement by a corresponding one of stems **592**.

Referring to FIG. **11**A, an unaligned position of assembly **558** is illustrated that shows teeth **566** and **576** of wheels **560** and **570**, respectively, out of register similar to the embodiment illustrated in FIG. **4**. Referring additionally to FIG. **11**B, a side elevational view of assembly **558** in the unaligned configuration is illustrated. Splines **561** of wheel

560 engage splines 571 of wheel 570. For each device 590*a*, 590*b*, stems 592 have corresponding longitudinal stem axes S1, S2. Stems 592 are inserted through corresponding washers 596 and apertures 596 to initially engage a corresponding threaded recess 579. Springs 580 are not substantially compressed in the configuration of FIGS. 11A and 11B.

Referring additionally to FIGS. 12A and 12B, a perspective view and a side elevational view of assembly 558 in an aligned configuration are illustrated, respectively. This aligned configuration generally corresponds to the aligned configuration of assembly 58 illustrated in FIG. 5. To provide alignment of assembly 558, stems 592 of adjustment devices 590*a*, 590*b* are further threaded into recesses 579 to compress springs 580 between wheel 560 and 570. As springs 580 are compressed, the inclines of mating splines 561, 571 provide a ramping action that generally converts the translational motion of devices 590*a*, 590*b* to a rotational motion of wheels 560, 570. As stems 592 of devices 590*a*, 590*b* are unthreaded, the compressed springs 580 provide a force which rotates wheels 560 and 570 in the opposite direction due to the engagement of splines 561, 571. Assembly 558 is configured so that teeth 566 and 576 are generally aligned when stems 592 are fully threaded into recesses 579. This aligned orientation of assembly 558 is also preferably configured to provide a selected maximum bias torque. The distance wheels 560 and 570 occupy along stem axes S1 and S2 changes from D1 for the unaligned position shown in FIG. 11B to D2 for the aligned position illustrated in FIG. 12B, where D1 is greater than D2. Notably, D2 is the minimum distance occupied by wheels 560, 570 of assembly 558 along stem axes S1, S2. Thus, wheels 560, 570 rotate relative to each other in accordance with the distance occupied by the wheels 560, 570 along the rotational axis corresponding to center 554.

Preferably the number of teeth 566 is the same as the number of teeth 576. It is also preferred that the number of helical splines 561, 571 be the same as the number of teeth 566, 576, respectively. Identical quantities of teeth and splines simplifies assembly by avoiding the need to index splines 571, 561 to assure that alignment of teeth 566 and 576 coincides with high spring compression. In other embodiments, aperture 569 may be configured as a non-circular opening as opposed to the generally circular opening illustrated in FIG. 10. In one alternative embodiment, aperture 569 is configured as an arcuate slot with a bend radius extending from center 554.

Splines 561, 571 may be provided in different locations besides hubs 563, 573. By way of non-limiting example, arcuate slots defined by one wheel may have an inner surface defining splines configured to mate with splines defined by a flange extending from the other wheel into these slots. Notably, one or more segments of mating spines oriented about the rotational axis are capable of providing the relative rotation of the gear wheels without needing to encircle the axis.

Similar to the embodiment of assembly 58, assembly 558 provides an alignment device which provides for selectively aligning teeth of two gear wheels of an anti-lash gear assembly by opposing the spring bias of the assembly. Stems 592 are tightened down to provide the aligned configuration of FIGS. 12A and 12B for installation. Once assembly 558 is meshed with another gear, gear 42 for example, stem 592 of each device 590*a*, 590*b* is loosened to permit relative rotation of wheels 560 and 570 to take-up lash of the mating gear. This loosened position would appear similar to the configuration of FIGS. 11A and 11B, but would preferably provide clearance between head 594 and washer 596 of each bolt 590 to accommodate changing lash conditions of a corresponding mesh. In one embodiment, devices 590*a*, 590*b* are removed once assembly 558 is installed in a mesh with another gear. This embodiment relies on the mesh to oppose the bias.

Each assembly 58, 558 is configured with an adjustment device having a threaded stem coupled to one wheel that extends along a stem axis. These devices further include a head coupled to the stem and configured for adjustable positioning relative to the wheel. Generally, assemblies 58 and 558 may be configured to be interchangeable with regard to other features of the present invention. Furthermore, assembly 58 or 558 may be adapted for use with anti-lash gear 200. In other embodiments of assemblies 58, 558; bolts 90, 590 may be replaced by a threaded stem fixed to one of the wheels with a nut threaded thereon to provide a movable head. This nut is positioned along the stem to selectively engage the other wheel. In still other embodiments of the present invention, neither anti-lash assembly may be utilized. Indeed in some alternative embodiments of the present invention, a conventional scissor gear assembly may be employed.

FIGS. 13 and 14 depict adjustable idler gear mechanism 600*a* of the present invention. Mechanism 600*a* is illustrated as a substitute for idler gear 100 and mechanism 120 of system 20 with like reference numerals representing like features. Mechanism 600*a* includes gear 603*a*, mounting shaft 655*a*, and mounting plate 630*a* coupled to block 22 as previously described in connection with system 20. Gear 603*a* includes gear wheel 613*a* and mounting shaft 655*a*. Gear wheel 613*a* has rim 616*a* defining teeth 615*a*. Rim 616*a* is integrally connected to web 617*a*. Web 617*a* is integrally connected to hub 618*a* which engages cylindrical bushing 619*a* to provide a rotational bearing relationship in the manner as previously described for idler gear 100.

Mounting shaft 655*a* has positioning pin 601*a* protruding from mounting surface 667*a*. Pin 601*a* is shown engaging slot 602*a* defined by mounting surface 637*a* of mounting plate 630*a*. Mounting surfaces 667*a* and 637*a* are also shown in engagement. Pin 601*a* is fixed relative to shaft 655*a* and is generally centered relative to the rotational axis that coincides with rotational center 650*a* of gear 603*a*. Pin 601*a* may be an integral part of shaft 655*a* or connected thereto using any suitable means as would occur to those skilled in the art. Slot 602*a* follows a limited pathway represented by a line segment with the reference numeral 660*a* in FIG. 14. Collectively, pin 601*a* and slot 602*a* provide a guide 625*a* to position center 650*a* along pathway 660*a*. 660*a* may follow a generally straight path, as represented by the solid line segment, or alternatively, it may follow a curvilinear path 660*a*' as shown in exaggerated form by the dashed line segment of FIG. 14. Plate 630*a* is fixed relative to block 22 using any suitable means that would occur to those skilled in the art. The location of plate 630*a* is selected to provide a predetermined relationship of pathway 660*a*, 660*a*' relative to rotational center 54 of gear 50.

In one preferred assembly process, rotational centers 54 and 204 of gears 50 and 200, respectively, are predetermined. Mounting plate 630*a* is positioned to establish that the distance separating rotational center 54 from any given point along pathway 660*a* stays within a predetermined range. This range corresponds to an acceptable range of backlash variation between gears 50 and 603*a* as more fully described hereinafter. After gears 50 and 200 are mounted, gear 603*a* is positioned to adjustably intermesh therewith and pin 601*a* is received in slot 602*a*. Pin 601*a* slidingly engages slot **602**a to provide guide **625**a, and thereby facilitate adjustment of rotational center **650**a of gear **603**a along pathway **660**a. For any position of pin **601**a along slot **602**a, the spacing between rotational centers **54** and **650**a is maintained to provide an acceptable minimal degree of backlash for the mesh between gears **603**a and **50** in accordance with the predetermined spatial relationship between pathway **660**a and rotational center **54**. For the generally straight pathway configuration designated by reference numeral **660**a, it is preferred to orient pathway **660**a to be approximately parallel to tangent line **610**a; where line **610**a is tangent to pitch circle **612**a for gear **50** and pitch circle **614**a for gear **603**a. This orientation generally minimizes the amount of backlash variation between gear **50** and gear **603**a as pin **601**a is moved along pathway **660**a.

Further, this technique of determining the position of assembly **600**a relative to gears **50, 200** is less prone to error, being more "foolproof" than existing methods to minimize backlash of two or more meshes of the same gear. Also, the structure of guide **625**a is less complicated and time consuming to adjust compared to conventional approaches.

Typically, it is easier to manufacture guide **625**a to define a generally straight pathway **660**a than a predetermined curved pathway **660**a'; however, the degree of accuracy available when matching the backlash of the meshes of both gears **50** and **200** may be generally less for pathway **660**a than curved path **660**a'. Thus, in an alternative preferred embodiment, slot **602**a follows curved pathway **660**a'. The curvature of pathway **660**a' is preferably selected to provide generally constant spacing from rotational center **54** as pin **601**a is moved therealong. For this preferred embodiment, pathway **660**a' has a generally constant radius of curvature originating from rotational center **54**, such that pathway **660**a' is a concentric arc relative to pitch circle **612**a of gear **50**.

Pathways **660**a and **660**a' are also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear **200**. Therefore, pin **601**a may be moved along pathway **660**a or **660**a' to determine a position which minimizes backlash between gear **200** and **603**a while not exceeding a desired level of backlash between gear **50** and **603**a. Once a position is selected, shaft **655**a is fixed relative to mounting plate **630**a and block **22**. This fixation may be accomplished with one or more fasteners, such as with bolts, screws, or rivets; a bonding process, such as soldering, brazing, welding, or adhesively joining; or through such other means as would occur to one skilled in the art. In another embodiment, pin **601**a protrudes from the plate **630**a and the slot **602**a is defined by the shaft **655**a to provide guide **625**a with comparable adjustment capability.

FIGS. **15** and **16** depict another idler gear adjustment mechanism **600**b of the present invention. Mechanism **600**b is illustrated as a substitute for idler gear **100** and mechanism **120** of system **20** with like reference numerals representing like features. Mechanism **600**b includes gear **603**b, mounting shaft **655**b, and mounting plate **630**b coupled to block **22** as previously described in connection with system **20**. Gear **603**b includes gear wheel **613**a, as described in connection with mechanism **600**a. Likewise, gear **603**b includes cylindrical bushing **619**b to provide a rotational bearing relationship between gear wheel **613**a and shaft **655**b.

Adjustment mechanism **600**b has a guide **625**b including positioning pins **601**b and slot **609**b. Mounting shaft **655**b has two positioning pins **601**b protruding from mounting surface **667**b. Pins **601**b are shown engaging slot **609**b defined by mounting surface **637**b of mounting plate **630**b. Mounting surfaces **667**b and **637**b are also shown in engagement. Pins **601**b are fixed relative to shaft **655**b in an off-center relationship relative to the rotational axis that coincides with rotational center **650**b of gear **603**b. Pins **601**b may be an integral part of shaft **655**b or connected thereto using any suitable means as would occur to those skilled in the art. Slot **609**b follows a limited pathway represented by a line segment with the reference numeral **660**b in FIG. **16**. Plate **630**b is fixed relative to block **22** using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, plate **630**b is mounted to block **22** to establish a predetermined spatial relationship between pathway **660**b and rotational center **54** of gear **50**. Gear **603**b is mounted to adjustably intermesh with gears **50** and **200** by slidingly engaging pins **601**b in slot **609**b. For any position of pins **601**b along slot **609**b, the spacing between rotational centers **54** and **650**b is maintained to provide an acceptable minimal degree of backlash for the mesh between gears **603**b and **50** in accordance with the predetermined spatial relationship between pathway **660**b and rotational center **54**. For the illustrated embodiment, pathway **660**b is represented by a generally straight segment that is generally parallel to tangent line **610**b; where line **610**b is tangent to pitch circle **612**b for gear **50** and pitch circle **614**b for gear **603**b. In an alternative embodiment, pathway **660**b' is represented by a dashed line, and is curved and configured in the manner described in connection with pathway **660**a' of mechanism **600**a. The curvature is exaggerated to more clearly depict pathway **660**b'.

Pathway **660**b is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear **200**. Therefore, pins **601**b may be moved along pathway **660**b to determine a position which minimizes backlash between gear **200** and **603**b while not exceeding a desired level of backlash between gears **50** and **603**b. Once a position is selected, shaft **655**b is fixed relative to mounting plate **630**b and block **22** in a manner comparable to that described for plate **630**a.

FIGS. **17** and **18** depict another idler gear adjustment mechanism **600**c of the present invention. Mechanism **600**c is illustrated as a substitute for idler gear **100** and mechanism **120** of system **20** with like reference numerals representing like features. Mechanism **600**c includes gear **603**c, mounting shaft **655**c, and mounting plate **630**c that have been coupled to block **22** previously described in connection with system **20**. Gear **603**c includes gear wheel **613**a described in connection with mechanism **600**a. Likewise, gear **603**c includes cylindrical bushing **619**c to provide a rotational bearing relationship between gear wheel **613**c and shaft **655**c.

Adjustment mechanism **600**c has a guide **625**c including pins **601**c and shoulder **604**c. Mounting plate **630**c has two positioning pins **601**c protruding from mounting surface **637**c. Pins **601**c are shown engaging guide shoulder **604**c defined by mounting surface **667**c of shaft **655**c. Mounting surfaces **667**c and **637**c are also shown in engagement. Pins **601**c are fixed relative to plate **630**c in an off-center relationship relative to the rotational axis that coincides with rotational center **650**c of gear **603**c. Pins **601**c may be an integral part of plate **630**c or connected thereto using any suitable means as would occur to those skilled in the art. Shoulder **604**c follows a pathway represented by a line segment with the reference numeral **660**c in FIG. **18**. Plate **630**c is fixed relative to block **22** using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, plate **630**c is mounted to block **22** to establish a predetermined spatial relationship between pathway **660***c* and rotational center **54** of gear **50**. Gear **603***c* is mounted to adjustably intermesh with gears **50** and **200** by sliding pins **601***c* against shoulder **604***c*. For any position of pins **601***c* along shoulder **604***c*, the spacing between rotational centers **54** and **650***c* is maintained to provide an acceptable minimal degree of backlash for the mesh between gears **603***c* and **50** in accordance with the predetermined spatial relationship between pathway **660***c* and rotational center **54**. For the illustrated embodiment, pathway **660***c* is represented by a generally straight segment that is generally parallel to tangent line **610***c*; where line **610***c* is tangent to pitch circle **612***c* for gear **50** and pitch circle **614***c* for gear **603***c*. In an alternative embodiment, curved pathway **660***c'* may be utilized as represented by the dashed line in FIG. **18**. Preferably, pathway **660***c'* is curved in the manner described for pathway **660***a'* of the mechanism **600***a*. The curvature of pathway **660***c'* is exaggerated to more clearly depict it in FIG. **18**.

Pathway **660***c* is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear **200**. Therefore, pins **601***c* may be moved along pathway **660***c* to determine a position which minimizes backlash between gear **200** and gear **603***c* while not exceeding a desired level of backlash between gear **50** and gear **603***c*. Once a position is selected, shaft **655***c* is fixed relative to mounting plate **630***c* and block **22** in a manner comparable to that described for plate **630***a*.

FIGS. **19** and **20** depict another idler gear adjustment mechanism **600***d* of the present invention. Mechanism **600***d* is illustrated as a substitute for idler gear **100** and mechanism **120** of system **20** with like reference numerals representing like features. Mechanism **600***d* includes gear **603***d*, mounting shaft **655***d*, and mounting plate **630***d* that have been coupled to block **22** previously described in connection with system **20**. Gear **603***d* includes gear wheel **613***a* described in connection with mechanism **600***a*. Likewise, gear **603***d* includes cylindrical bushing **619***d* to provide a rotational bearing relationship between gear wheel **613***a* and shaft **655***d*.

Adjustment mechanism **600***d* has a guide **625***d* including positioning pins **601***d* and guide walls **604***d*. Mounting plate **630***d* has two positioning pins **601***d* projecting from mounting surface **637***d* that are oppositely disposed relative to rotational center **650***d*. Pins **601***d* each engage a corresponding one of a pair of guide walls **604***d* defined by surface **667***d* of shaft **655***d*. Guide walls **604***d* are generally parallel to each other and are generally opposite each other. Surfaces **667***d* and **637***d* are also shown in engagement. Pins **601***d* are fixed relative to plate **630***d* in a generally symmetric relationship about rotational center **650***d* of gear **603***d*. Pins **601***d* may be an integral part of plate **630***d* or connected thereto using any suitable means as would occur to those skilled in the art. Walls **604***d* generally correspond to a pathway represented by a line segment with the reference numeral **660***d* in FIG. **20**. Plate **630***d* is fixed relative to block **22** using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, plate **630***d* is mounted to block **22** to establish a predetermined spatial relationship between pathway **660***d* and rotational center **54** of gear **50**. Gear **603***d* is mounted to adjustably mesh with gears **50** and **200** by sliding each pin **601***d* against a corresponding wall **604***d*. For any position of pins **601***d* along walls **604***d*, the spacing between rotational centers **54** and **650***d* is maintained to provide an acceptable minimal degree of backlash for the mesh between gears **603***d* and **50** in accordance with the predetermined spatial relationship between pathway **660***d* and rotational center **54**. For the illustrated embodiment, pathway **660***d* is represented by a generally straight segment that is generally parallel to tangent line **610***d*; where line **610***d* is tangent to pitch circle **612***d* for gear **50** and pitch circle **614***d* for gear **603***d*. In an alternative embodiment, the pathway is represented by a dashed line in FIG. **20** with reference numeral **660***d'* and is curved in the manner described in connection with pathway **660***a'* of mechanism **600***a*. The curvature of pathway **660***d'* is exaggerated in FIG. **20** to more clearly depict it.

Pathway **660***d* is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear **200**. Therefore, pins **601***d* may be moved along pathway **660***d* to determine a position which minimizes backlash between gear **200** and gear **603***d* while not exceeding a desired level of backlash between gear **50** and gear **603***d*. Once a position is selected, shaft **655***d* is fixed relative to mounting plate **630***d* and block **22** in a manner comparable to that described for plate **630***a*.

FIGS. **21** and **22** depict another idler gear adjustment mechanism **600***e* of the present invention. Mechanism **600***e* is illustrated as a substitute for idler gear **100** and mechanism **120** of system **20** with like reference numerals representing like features. Mechanism **600***e* includes gear **603***e*, mounting shaft **655***e*, and mounting rail **630***e* coupled to block **22** as previously described in connection with system **20**. Gear **603***e* includes gear wheel **613***a* as described in connection with mechanism **600***a*. Likewise, gear **603***e* includes cylindrical bushing **619***e* to provide a rotational bearing relationship between gear wheel **613***a* and shaft **655***e*.

Adjustment mechanism **600***e* has guide **625***e* including a channel **605***e* and rail **630***e*. Shaft **655***e* has mounting surface **667***e* that defines channel **605***e*. Channel **605***e* receives rail **630***e* therein, engaging mounting surface **637***e* of rail **630***e* with surface **667***e*. Channel **605***e* generally corresponds to a pathway represented by a line segment with the reference numeral **660***e* in FIG. **22**. Rail **630***e* is fixed relative to block **22** using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, rail **630***e* is mounted to establish a predetermined spatial relationship between pathway **660***e* and rotational center **54** of gear **50**. Gear **603***e* is adjustably mounted to mesh with gears **50** and **200** by slidingly engaging rail **630***e* in channel **605***e*. For any position of rail **630***e* in channel **605***e*, the spacing between rotational centers **54** and **650***e* is maintained to provide an acceptable minimal degree of backlash for the mesh between gears **603***e* and **50** in accordance with the predetermined spatial relationship between pathway **660***e* and rotational center **54**. For the illustrated embodiment, pathway **660***e* is represented by a generally straight segment that is generally parallel to tangent line **610***e*; where line **610***e* is tangent to pitch circle **612***e* for gear **50** and pitch circle **614***e* for gear **603***e*. In an alternative embodiment, the pathway is represented by the dashed line in FIG. **22** with reference numeral **660***e'*; and is curved in the manner described in connection with pathway **660***a'* of mechanism **600***a*. The curvature of pathway **660***e'* is exaggerated in FIG. **22** to more clearly depict it.

Pathway **660***e* is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear **200**. Therefore, rail **630***e* may be moved along channel **605***e* to determine a position which minimizes backlash between gear **200** and gear **603***e* while not exceeding a desired level of backlash between gear **50** and gear **603***e*. Once a position is selected, shaft

655*e* is fixed relative to rail 630*e* and block 22 in a manner comparable to that described for plate 630*a*.

FIGS. 23 and 24 depict another idler gear adjustment mechanism 600*f* of the present invention. Mechanism 600*f* is illustrated as a substitute for idler gear 100 and mechanism 120 of system 20 with like reference numerals representing like features. Mechanism 600*f* includes gear 603*f*, mounting shaft 655*f*, and guide plate 630*f* that have been coupled to block 22 previously described in connection with system 20. Gear 603*f* includes gear wheel 613*a* described in connection with mechanism 600*a*. Likewise, gear 603*f* includes cylindrical bushing 619*f* to provide a rotational bearing relationship between gear wheel 613*a* and shaft 655*f*.

Shaft 655*f* has mounting surface 607*f* engaging a cheek 606*f* of plate 630*f*. Cheek 606*f* generally extends along a pathway represented by a line segment with the reference numeral 660*f* in FIG. 24. Plate 630*f* is fixed relative to block 22 using any suitable means that would occur to those skilled in the art.

In one preferred assembly process, plate 630*f* is mounted to establish a predetermined spatial relationship between pathway 660*f* and rotational center 54 of gear 50. Gear 603*f* is adjustably mounted to mesh with gears 50 and 200 by engaging mounting surface 607*f* of shaft 655*f* against cheek 606*f* in a sliding or rolling relationship. For any position of shaft 655*f* along cheek 606*f*, the spacing between rotational centers 54 and 650*f* is maintained to provide an acceptable minimal degree of backlash for the mesh between gears 603*f* and 50 in accordance with the predetermined spatial relationship between pathway 660*f* and rotational center 54. For the illustrated embodiment, pathway 660*f* is represented by a generally straight segment that is generally parallel to tangent line 610*f*; where line 610*f* is tangent to pitch circle 612*f* for gear 50 and pitch circle 614*f* for gear 603*f*. In an alternative embodiment, the pathway is represented by a dashed line in FIG. 24 with reference numeral 660*f*'; and is curved in the manner described in connection with pathway 660*a*' of mechanism 600*a*. The curvature of pathway 660*f*' is exaggerated in FIG. 24 to more clearly depict it.

Pathway 660*f* is also oriented to provide a substantially greater degree of spacing variation and corresponding backlash variation with respect to gear 200. Therefore, shaft 655*f* may be moved along cheek 606*f* to determine a position which minimizes backlash between gear 200 and gear 603*f* while not exceeding a desired level of backlash between gear 50 and gear 603*f*. Once a position is selected, shaft 655*f* is fixed relative to plate 630*f* and block 22 in a manner comparable to that described for plate 630*a*.

It should be understood that additional gears and different gear types may be used with mechanisms 600*a*, 600*b*, 600*c*, 600*d*, 600*e*, 600*f* (collectively designated mechanisms 600) as would occur to those skilled in the art. Moreover, various guide pathways may be established relative to center 204 instead of 54 with a corresponding change in backlash adjustability. Further, it should be appreciated that the various pins of mechanisms 600 may be joined by a bonding process, such as soldering, brazing, welding; by adhesive application; by threading, by a press-fit connection, or through such other means as would occur to those skilled in the art. The pins are preferably cylindrical in shape, but other embodiments may include pins with a different shape. Also various other guide members, such as slots, shoulders, walls, and channels, of mechanisms 600 may be machined, casted, or formed using techniques known to those skilled in the art. Preferably, all parts for mechanisms 600 are made from a metal or composite material suitable for heavy-duty diesel engine applications.

It should also be appreciated that each mechanism 600 facilitates assembly of a gear train with minimized backlash by constraining backlash variation between the idler gear and gear 54 to an acceptable level, and yet permitting generally independent adjustments to reduce backlash with gear 200. This feature makes it unnecessary to precision mount gear 200 and thus improve assembly effectiveness in terms of time and cost. Moreover, backlash reduction in accordance with the present invention generally provides a quieter, longer lasting gear train.

All publications, patents, and patent applications cited in this specification are herein incorporated by reference as if each individual publication, patent, or patent application was specifically and individually indicated to be incorporated by reference and set forth in its entirety herein.

While the invention has been illustrated and described in detail in the drawings and foregoing description, the same is to be considered as illustrative and not restrictive in character, it being understood that only the preferred embodiment has been shown and described and that all changes, modifications, and equivalents that come within the spirit of the invention as defined by the following claims are desired to be protected.

What is claimed is:

1. A gear train assembly for an engine, comprising:

(a) a first scissor gear rotatably coupled to the engine, said first scissor gear including a first rotational center;

(b) a second scissor gear rotatably coupled to the engine, said second scissor gear including a second rotational center; and

(c) an adjustable idler gear mechanism coupled to the engine, said mechanism including an idler gear forming a first mesh with said first scissor gear and a second mesh with said second scissor gear, said idler gear including a third rotational center, said mechanism further including a first pin fixed relative to said third rotational center and a guide surface slidably engaged by said pin to define a pathway to adjustably position said third rotational center, said pathway defining a predetermined spatial relationship with said first rotational center to keep backlash between said first scissor gear and said idler gear at a predetermined minimum for a range of positions of said third rotational center along said pathway, said range of positions further providing a range of backlash adjustment between said idler gear and said second scissor gear.

2. The assembly of claim 1, wherein said pathway is generally straight.

3. The assembly of claim 1, wherein said pathway is generally curvilinear.

4. The assembly of claim 3, wherein said pathway is in the form of an arc that is equidistant form said first rotational center.

5. The assembly of claim 1, wherein said mechanism defines a slot bounded by said guide surface.

6. The assembly of claim 5, wherein said mechanism includes a second pin engaging said slot.

7. The assembly of claim 6, further comprising means for selectively fixing the second pin relative to said guide surface.

8. A gear train assembly for an engine, comprising;

(a) a first scissor gear rotatably coupled to the engine, said first scissor gear including a first rotational center;

(b) a second scissor gear rotatably coupled to the engine, said second scissor gear including a second rotational center; and

(c) an adjustable idler gear mechanism coupled to the engine, said mechanism including an idler gear forming a first mesh with said first scissor gear and a second mesh with said second scissor gear, said idler gear including a third rotational center, said mechanism including a channel and a rail slidably engaging said channel to adjust said third rotational center therealong, said rail defining a predetermined spatial relationship with said first rotational center to keep backlash between said first scissor gear and said idler gear at a predetermined minimum for a range of positions of said third rotational center along said rail, said range of positions further providing a range of backlash adjustment between said idler gear and said second scissor gear.

9. The assembly of claim **8**, wherein said channel defines a generally straight pathway.

10. The assembly of claim **8**, further comprising means for fixing said rail in said channel.

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